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Engineering

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ENGINEERED CLIMATE

Some Thoughts on Energy

THE quiet luster of the engineer reflects everywhere from today's evidences of technological advancement. Appreciation—notably as expressed by the nation's buying patterns—comes from a people who consume in energy some two-hundred-fifty times that available from their own muscles.

The engineer fits into this pattern of service to mankind through his training and practice in seeking always to wrest a maximum of benefit from the bounty of nature's raw materials and from available manpower. While concerned with methods and processes of manufacture, he develops the intermediate devices—among them internal combustion engines of both the internal spark ignition and Diesel types, Diesel-electric locomotives, jet engines, and electric motors—by which man extracts and molds more and more from nature. These intermediate devices operate on the fuels and energy of nature to provide an ever-increasing inventory of physical good things for more people.

Mechanization is apparent wherever one looks. Mechanization—an expression of technological advancement—begets mechanization. It is as though the engineer were casting stones of progress onto still waters. The waves form additive and expanding circles to extend into areas of good not before touched. Achievements invite still further achievements and, thus, industry grows—as the working force joins in actually making the desirable commodities. Our technology of the past half century is replete with examples of how engineers' dreams have transformed and multiplied the people's energy; the best known, perhaps, is the transformation of our transportation system from largely animal power to machine power.

The engineer conventionally works behind the scenes—for his work cannot become of actual value until a host of others have contributed in major and minor ways to his effort. The engineer, by applying technical fundamentals, improves existing products, methods, and processes and, thus, adds depth to his company's contribution to society. By reaching into the unknown through experimentation and application of the scientific approach, he develops still newer products, methods, and processes and, thus, lengthens the base from which his organization can serve.

Because the engineer's handiwork reflects from so many areas of today's industry, he frequently joins, too, in the long-range planning and guidance of his organization—even in non-technical areas. The same disciplined thinking which can bring the physically new can apply to non-technical areas of management. Today's decisions are largely non-technical but—because we live in a technological age generated by technical minds—a large percentage of the decisions are strengthened by a knowledge of the technical procedures involved.

Thus, by adding depth and length to his organization, the engineer also can train himself to be valuable in adding stature to his organization. In the first two areas, he is indispensable; in the latter area he can become valuable in accordance with his ability to sense and lead the still-increasing contribution of engineering—through industry—to the people.

Cyrus R. Osborn,
Vice President in Charge
of Engine Group
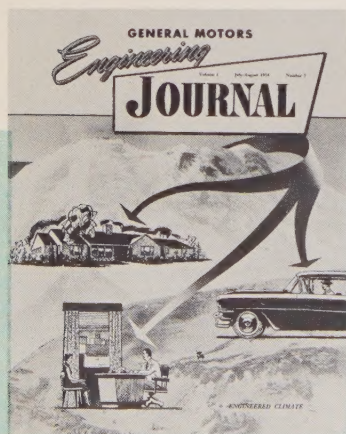
THE COVER

The cool, bracing air of a mountain breeze is suggested in the cover design by Artist John Tabb. Man's normal activities are symbolized by his home, his place of business, and his automobile. Fortunately, he can bring into a great many of his activities, regardless of locality, air having a mountain-quality freshness. This he can do because of the application of sound engineering principles. Air conditioning supplies crisp, clean air into his environment providing a new background for more comfortable living. Forward-looking designs and advanced manufacturing techniques are demonstrating daily that the marvel of air conditioning fits almost everywhere into man's existence.

CONTENTS

An Approach to the Design and Development of a New Product: A Light-Weight, Two-Cycle Diesel Engine

By ERIC R. BRATER

Cleveland Diesel

Engine Division

The task was to design and develop a Diesel engine for military use which would be more powerful, lighter, and smaller than existing conventional engines. Specifications in the contract which came to Cleveland Diesel Engine Division established certain size, proportion, weight, and performance standards. Knowledge and practice gained from previous successful completion of similar projects gave engineers a starting point. Yet, nothing was taken for granted. With their goal set as a successful product which could be manufactured with maximum ease, engineers—including specialists in design, mathematics, metallurgy, torsional vibration, instrumentation, testing, lubrication, and other fields—arrived at a tentative design. These human contributions resulted in a long series of experimental expedients, such as wood and aluminum models, which were subjected to considerable study and developmental testing, and changes were made as test results indicated. When the production model was achieved and accepted, final testing and field experience proved that the application of well-known techniques combined with study and experimental analysis of each new engine part resulted in a new engine which met all the desired requirements.

A case example in how theoretical and practical methods are combined

To satisfy demands for Diesel engines with more power, less weight, and occupying less space than conventional engines, the Cleveland Diesel Engine Division of General Motors embarked on an elaborate and extensive design and development program to produce a Diesel engine which would meet the exacting and severe requirements of specialized application in connection with the national defense.

It was recognized that, for such an engine, not only the best known materials had to be used but also that these materials had to be applied most effectively to withstand high stresses and pressures far in excess of those encountered in most existing designs. Each engine part was subjected to the most thorough engineering analysis known, to obtain greatest lightness and yet ample strength.

To accomplish this, the most modern means of instrumentation techniques and experimental stress analyses were enlisted to guide the designers during the various stages of the developmental work.

To reduce the weight per horsepower to about one-half that of Diesel engines used previously for the same application, steel construction was used throughout. To satisfy extremely rigid shock-resistance specifications, the use of cast iron and aluminum was held to a minimum.

After a thorough preliminary investigation of this problem, keeping space limitations in mind, it was concluded that a vertical, radial, 16-cylinder, two-cycle Diesel engine would most advantageously answer the purpose (Fig. 1).

Fitting the Task into the Organization

The first step which had to be taken was to prepare the Engineering Department's working schedule consisting of design, layout work, detailing, checking, and engineering releases of the experimental pilot engine to Purchasing and Manufacturing Departments. Further, it was necessary to schedule tooling, routing of parts in the plant, setup for experimental testing, release of engine for production, completion of production engines, and shipment to the user. This all had to be carefully worked out and schedules maintained.

To execute the design and construction successfully, a competent, well-trained, and coordinated engineering organization was necessary. Many years of experience have indicated that it is highly advantageous to concentrate the technical and scientific work in a group of specialists, each of whom are experts in certain phases of engineering, such as vibration, electronics, instrumentation, mathematical calculations, development testing, and in specialized projects. This technical group worked in close coordination with the designers who carried out the actual layout work.
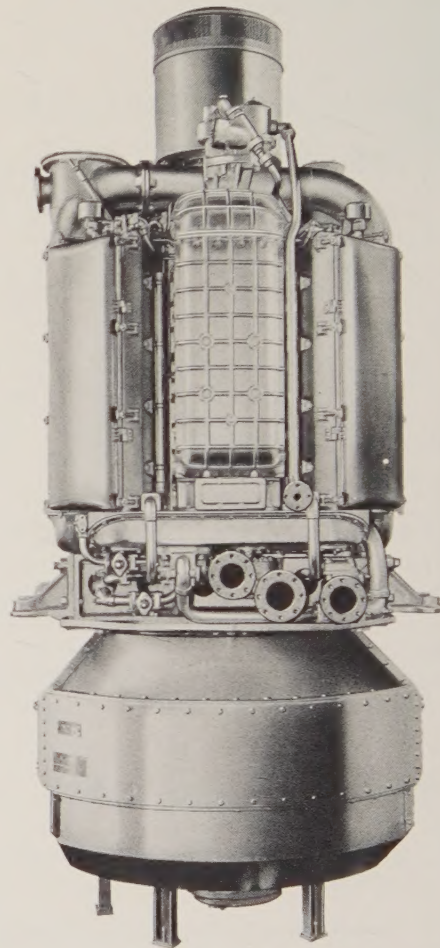
Fig. 1—Before this final Diesel-engine design could be achieved, thousands of interrelated technical decisions had to be reached. Engineers determined early in their work that the new engine should be of all-steel, vertical, radial, 16-cylinder, two-cycle design. Then, the design was made practical, manufacturable, and appropriately related in every way to its generator and intended military use.

Preliminary Design Decisions

After the engine shape and number of cylinders were established, it became necessary to consider crank arrangement
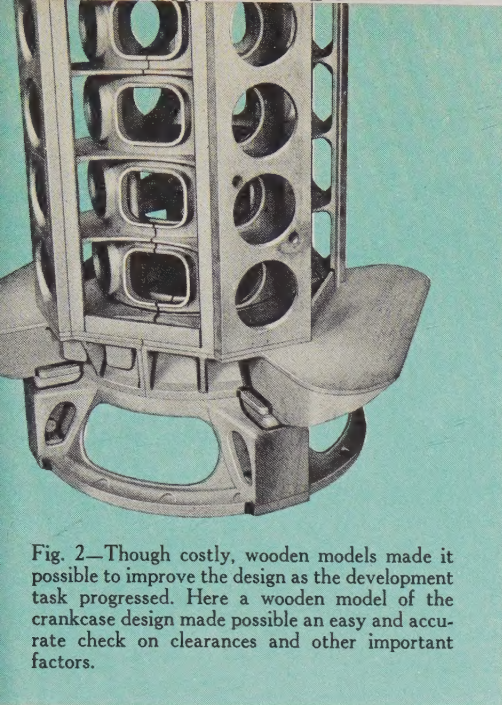
Fig. 2—Though costly, wooden models made it possible to improve the design as the development task progressed. Here a wooden model of the crankcase design made possible an easy and accurate check on clearances and other important factors.
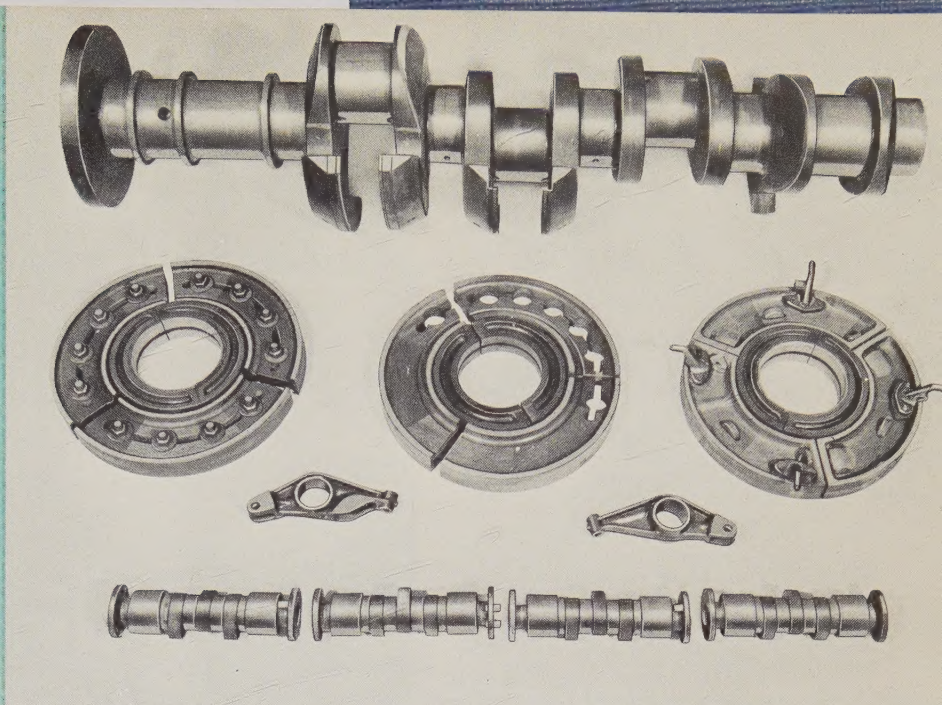
Fig. 3—Among the many parts for which wooden models were made for study were (top to bottom) a crankshaft, three different main-bearing designs, exhaust and injector rocker levers, and a camshaft in four sections. The models helped to determine how final designs could be improved.

and firing order from a balance and torsional-vibration standpoint. This is of extreme importance since torsional critical speeds, if occurring in the operating range, are difficult to minimize or eliminate after the engine is completed.

Calculations indicated that a bore of 6 in. and a stroke of 6½ in. would give the engine the required compactness radially, which permitted the design of a sturdy crankshaft with rigid crank-cheeks and ample overlap between the crankpin and the main journals.

The engine is of the uniflow scavenging-type arranged with four rows of four cylinders each and with 90° spacing between the banks. Four exhaust valves are located in the cylinder head. Air is supplied by four Roots-type positive displacement blowers. The air-intake ports are arranged radially in the liner.

A vast number of calculations were necessary to determine the proportions, shape, and strength of the various engine parts, such as the crankshaft, connecting rods, pistons, exhaust valves, cams, blowers, air and exhaust passages, and the water, lubricating-oil, and fuel-oil systems. The determination of exhaust, air-intake, and injector timing was based on well-established previous practice, modified to satisfy high-speed engine requirements. The calculated stresses, pressures, allowable deflections, and factors of safety were based on the experience obtained with previously designed engines.

Engine-part proportions, based on ideal technical considerations, could not

always be realized and, therefore, a constant exchange of ideas and suggestions took place between the technical group and the designers until, after making numerous preliminary layouts, a realistic, practical solution was found.

Development Testing

Wooden Models

The first in a long series of experimental expedients was the making of wooden models which enabled engineers to arrive at a sound design, combining greatest strength and lightness of engine parts. The great value of such models cannot be over-emphasized. Although expensive, wooden models proved to pay good dividends. During the early design stages, there appeared to be no suitable substitute for wooden models to obtain a clear conception of what a part would look and feel like.

For example, careful study of a wooden model of the crankcase (Fig. 2) made it both possible and certain that corrections were made to undesirable design features which might have escaped the designers' attention on the layout—design features relative to proper proportions, fillets, clearances, and accessibility.

Wooden models were also built of all other vital parts, such as the crankshaft, three different designs of main bearings, exhaust and injector rocker levers, and

the camshaft made in four sections (Fig. 3).

Photoelastic Models

The next stage in developing the crankcase design and proportions was to analyze the main stress members by means of photoelastic models (Fig. 4). The highest stress concentrations were found around the oil-drainage holes. Experimental stress analyses made later indicated that this was a real problem since the stress concentrations were more severe than originally anticipated. Various models were made and tested with different sizes and shapes of oil-drainage holes until the optimum design was achieved.

Photoelastic studies, made with statically loaded plastic models, proved extremely helpful in discovering areas of high stress concentration. The plastic model, when viewed with polarized light, shows fringe patterns of the stress lines. A special transparent plastic was used for this experimentation.

Aluminum Models—Stresscoat Testing

During the early stages of the design, full-size experimental-crankcase main stress members were made of aluminum. These were set up and loaded by means of a hydraulic loading fixture. Then stresscoat or brittle lacquer was applied. The lacquer cracks when the part to
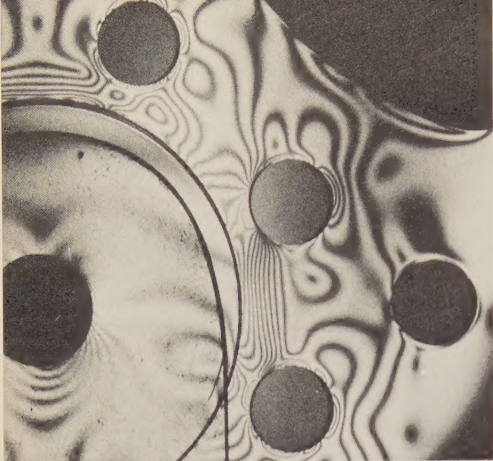
Fig. 4—For many parts, photoelastic models were made of a special transparent plastic and subjected to static loading. Here, part of a crankcase model as viewed with polarized light shows high stress-concentration lines between oil-drainage holes. In this case, the studies helped determine the best size and shape for these holes.
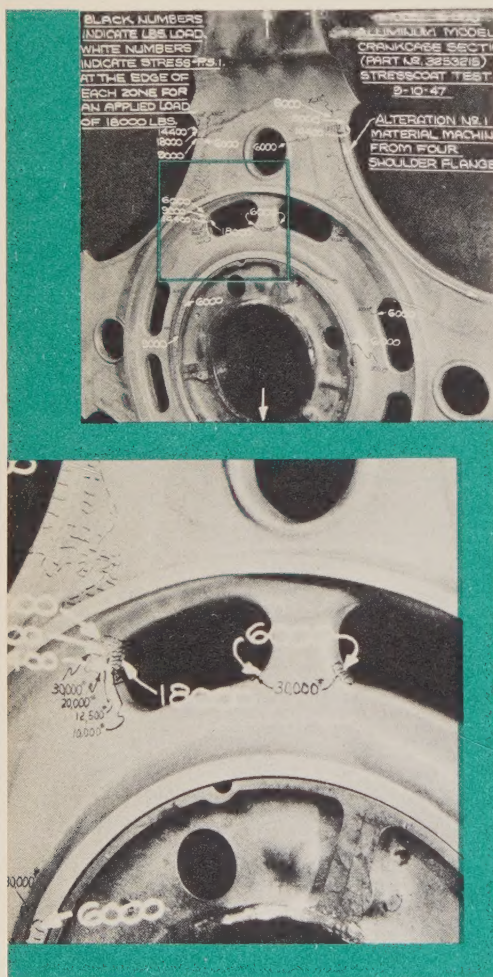
Fig. 5—Aluminum models provided a further opportunity to study stresses. Here, a crankcase section is labeled with data obtained from stresscoat or brittle-lacquer tests. The lacquer develops cracks when subjected to sufficient static loading, as the inset of the unretouched working photograph shows. White numbers indicate stress in psi and black numbers indicate the load in lb.
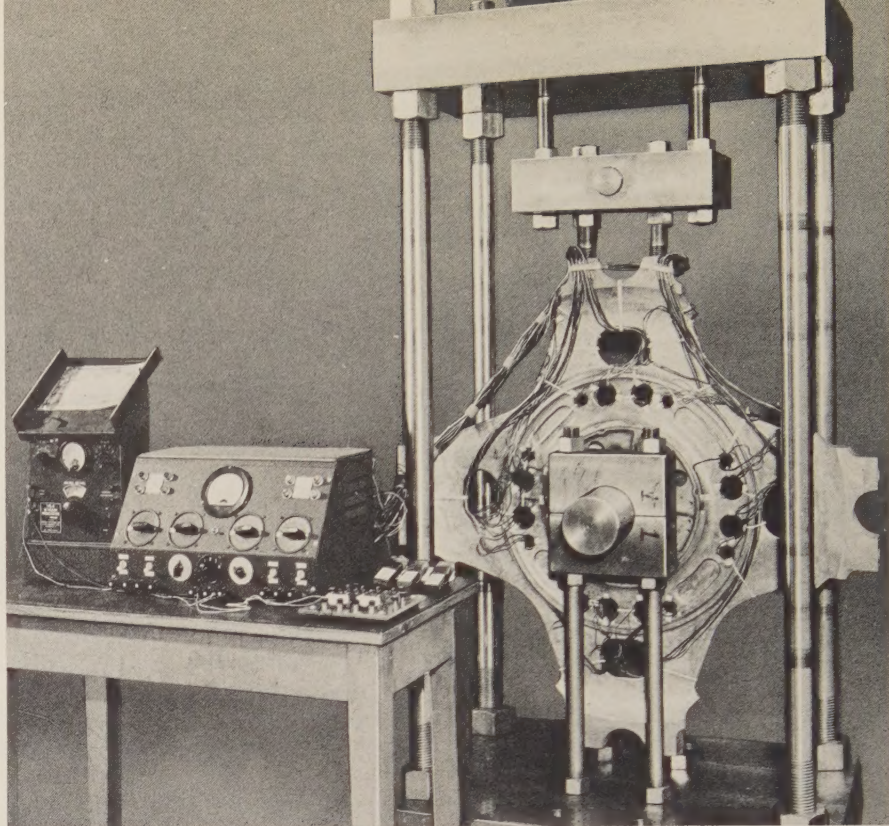
Fig. 6—Stresscoat and photoelastic-model testing gave good approximating data and indicated the best locations for individual strain gages in the crankcase-test setup shown here. Strain-gage data were considered most accurate but other data usually were found to be in good correlation.

be tested is loaded, indicating areas of maximum stress concentration. This was an extremely useful technique to precede a strain-gage stress analysis since it indicated where the bonded-wire strain gages should be placed. In a typical example of stresscoat testing of an aluminum model, the crankcase section shown in Fig. 5 was first loaded up to 10,000 lb, at which load cracks began to appear in the lacquer. Then the load was increased to 20,000 lb and finally to 30,000 lb. In this case the normal load for the final product was established at 18,000 lb, and test engineers were able to determine a satisfactory approximation of stress at various points on the crankcase by comparison with stress patterns on test strips loaded in a calibrating instrument furnished by the manufacturer of the stresscoat equipment used. Stresses determined by this method were found to be in good agreement with those subsequently determined by other means, such as strain-gage tests.

Next, about fifty strain gages were mounted in various locations on the main stress member (Fig. 6). The results obtained provided a more accurate stress check on the earlier stresscoat determi-

nations. As a result of stresscoat and strain-gage experimental analysis, many modifications were made to improve the design.

Bonded-wire strain gages are devices which can be obtained in various sizes for static and dynamic stress determination. They are cemented on the part to be tested. The fine wire embedded in the strain gage follows the strain in the part being tested; therefore, the wire's length changes, resulting in a change in its electrical resistance which can be measured. Then, by predetermined calibration methods, the change in resistance is translated to change in stress.

Bronze Models

Another example of an engine part which was analyzed is the main-bearing carrier. Extensive tests were performed to determine the stresses and deflections due to the tightening of section bolts during assembly and due to the crankshaft main-bearing loads. Experimental carriers were made of bronze and were subjected to stresscoat tests. The bearing load, when cracks first appeared in the lacquer, was 17,000 lb, resulting in an approximate stress of about 12,000 psi.
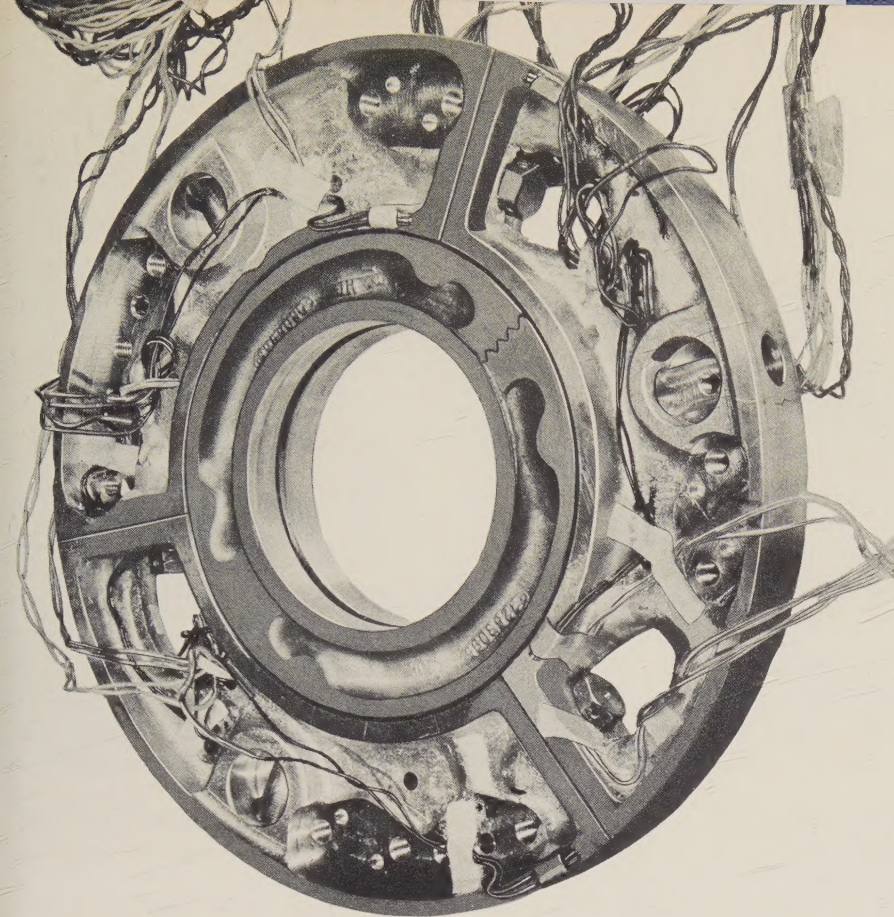
Fig. 7—Setup for a strain-gage experimental stress analysis of a bronze model of the main-bearing carrier. Results obtained enabled distortion to be reduced to a minimum through redesign.

Then about twenty-five strain gages were used for conventional experimental stress analysis of the part (Fig. 7). The increase in stress was observed while the carrier-section bolts were tightened uniformly. By proper design, based on the results obtained, distortion was reduced to a minimum.

First Steel Forging—Fatigue Testing

Finally, as soon as a steel forging was available, the main stress member was subjected to fatigue testing in a fatigue-testing machine at the normal calculated load of 18,000 lb for about 17 million cycles. After this severe test, the stress member was found to be in perfect condition, indicating the soundness of its design.

Interrelationship of Test Results

Summarizing the means used for experimental testing and stress analysis—wooden models, photoelastic models, stresscoat testing, strain-gage analysis, and fatigue testing—it is to be noted that each experiment filled a certain place in

the chain of sequence while developing an engine part. None of these tests told the whole story and, by themselves, they often might have been meaningless. Only the combined results presented a clear picture of the soundness of the design and proved that the material was used to the fullest advantage. A number of other engine parts, such as connecting rods, connecting-rod retainer rings, pistons, rocker levers, cylinder-head studs, and other vital parts were subjected to the same experimental procedure as was the main stress member of the crankcase.

Metallurgical Considerations

The assistance of an efficient Metallurgical Department, to select proper materials, to specify correct heat treatment, and to examine and analyze forgings and finished engine parts, contributed in no small way to the successful conclusion of the design, manufacture, and operation of the engine.

During the experimental testing of the engine, weaknesses showed up in certain parts which required improvement of physical properties, necessitating changes in material, surface hardness, or heat
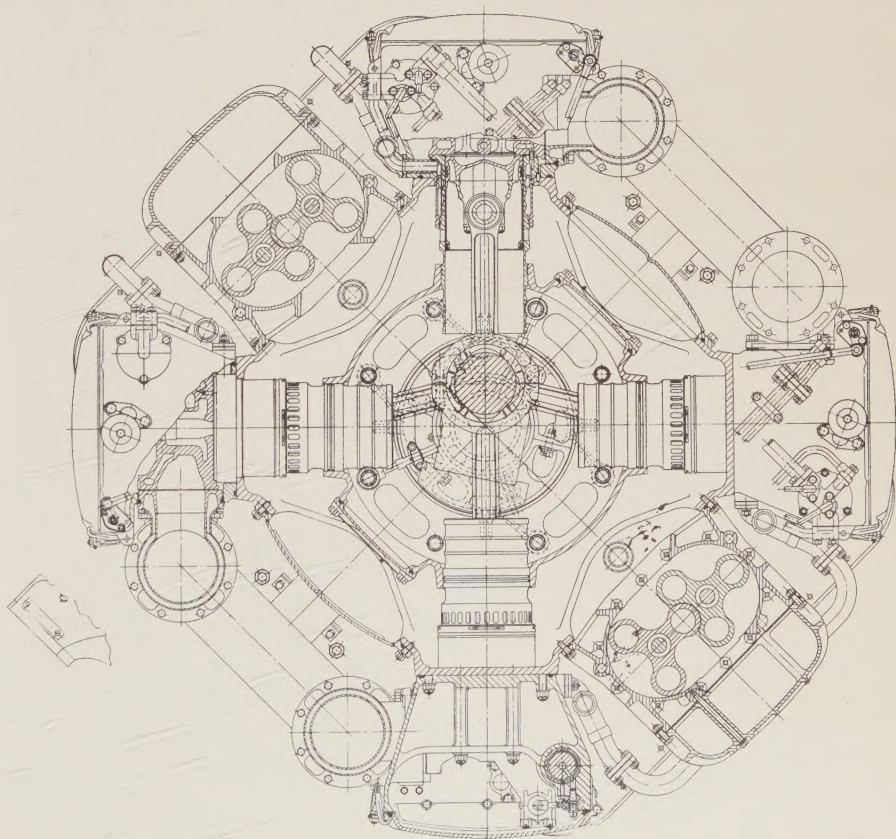
Fig. 8—Engine cross section showing one row of pistons, connecting rods, and cylinders, Roots-type blowers, and exhaust manifolds.
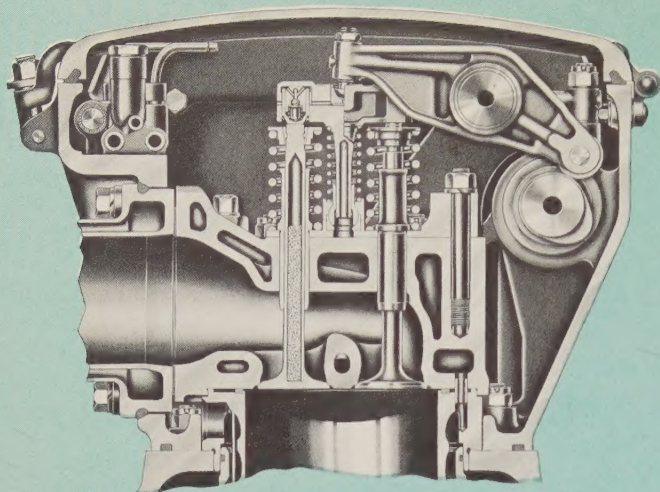
Fig. 9—Cross-sectional view through the cylinder head and exhaust valves.
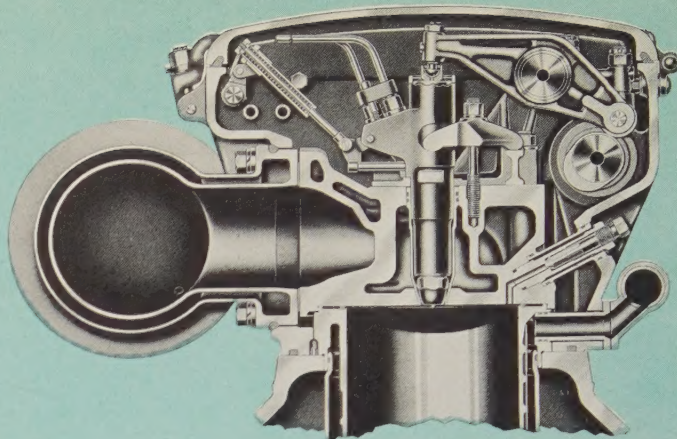
Fig. 10—Cross-sectional view through the cylinder head and the standard type General Motors unit injector with control linkage and rack. Also shown is the exhaust manifold (left) and the water-inlet manifold (right).

treatment. For instance, the gears in the accessory-drive gear train were made of S.A.E. 4340 chrome, nickel, molybdenum steel, heat-treated to a Brinell hardness of 311-352. It was found that this surface hardness was not sufficient to give long life, but additional nitriding of the gears, which increased the surface hardness to 85 Rockwell 15N scale (approximately 500 Vickers Brinell), solved the problem.

Overall Design— Manufacturing Considerations

The most painstaking engineering work may be worthless if parts are not manufactured properly with the greatest care to avoid stress raisers resulting from tool marks, improper fillets, or poor surface finish.

Stress concentrations may be doubled or tripled as a result of a sharp corner or too small a fillet. In this engine, therefore, such vital parts as connecting rods and connecting-rod retainer rings were machined all over and polished.

Fig. 8 is a cross section of the engine. It shows one row of pistons, connecting rods, and cylinders, Roots-type blowers, and exhaust manifolds. The four slipper-type connecting rods are acting on one crankpin and are located in the same plane. The forged-steel pistons are cooled by an oil spray through nozzles and also through the drilled connecting rod. The barrel-shaped crankcase is built up entirely of steel forgings welded together. The crankshaft is made of alloy steel, counterweighted for perfect engine balance. The ball-bearing mounted accessory gears are located at the bottom of the engine and drive the camshafts,

blowers, and water pumps. The gears also drive, through extension shafts, the air starting motor, governor, fuel pump, and tachometer, which are mounted on the top of the engine. The overspeed trip mechanism, which is of the flyweight-type, is mounted on the upper end of the crankshaft. In case the engine overspeeds, the air emergency shut-down valve located between the air intake and blower is closed by the overspeed trip mechanism which shuts off the air and stops the engine.

A section through the cylinder head and exhaust valves is shown in Fig. 9. The liquid-cooled exhaust valves are

Fig. 11—View of the finished barrel-shaped crankcase consisting of 67 individual steel forgings welded together automatically.

made of austenitic steel. Two exhaust valves are actuated from the overhead camshaft through one rocker lever, one valve bridge, and hydraulic lash adjusters.

Fig. 10 shows a section through the cylinder head and the standard type General Motors unit injector with rack and control linkage.

The exhaust valves open 89° before bottom center, and close 53° after bottom center. The air-intake ports are uncovered and covered 49° before and 49° after bottom center. At full load, the fuel is injected between 20° and 7° before top center, with zero fuel or no-load position at 14° before top center.

The barrel-shaped crankcase consists of 67 individual steel forgings welded together (Fig. 11). The first experimental crankcase was hand-welded. A mass-production method which is entirely automatic is now used to fabricate the crankcase.

The drop-forged alloy-steel crankshaft is counterweighted. Due to the crank arrangement it was impossible to balance each individual crankthrow 100 per cent but the crankshaft and engine are completely balanced. There are no unbalanced forces or couples. The crankshaft is also balanced statically and dynamically during manufacture. The counterweights are bolted on. Both main-bearing journals and crankpins are 4¾ in. in diameter.

One requirement of the engine construction was that it must be possible to remove and replace the main bearings without disturbing the crankshaft. This resulted in the design shown in Fig. 12. The main-bearing shells are steel backed

rods on the crankpin. Each retainer ring is made in two halves, serrated at the split line, and bolted together, and is free to rotate on the pads of the connecting rod.

Although in a two-cycle engine it is generally assumed that the force on the piston is always directed toward the crankshaft, at high speed there is a reversal in load which must be carried by the retaining rings. The ring bolts are made with right- and left-hand threads so that ring halves can be drawn together at the same time by turning both bolts simultaneously.

The connecting-rod retainer ring, being an extremely vital part of the engine, was also subjected to considerable experimental testing to determine the most advantageous shape and strength of the cross section, considering the limited space available. Both stresscoat and strain-gage tests were made. On the basis of these tests, and after making a number

Fig. 12—Main-bearing carrier consisting of three steel sections bolted together at the split lines and held in position in the crankcase by three adjustable dowels 5 and 7. One dowel 7, which is a snug fit in the crankcase, prevents turning and up and down movement of the main-bearing carrier. The other two dowels 5 fit in slots in the crankcase and prevent up and down movement only. After the carriers are located in the crankcase the dowels are secured by pin bolts 4. The screens shown in the bolt-clearance holes of each section prevent parts from falling through during assembly. The main-bearing shell 2 is made in two halves, serrated at the split line, and secured by dowels 3 to the carrier. Other identified parts are 1 carrier clamp bolt, 6 main-bearing shells, and 8 shell clamp bolt.

Fig. 13 (Right)—Slipper-type connecting rods were designed so that the segmental bearing shell 3 can be easily removed and replaced in service. Other identified elements are 1 connecting rod, 2 piston-pin bushing, 4 retainer rings, 5 ring bolts, 6 ring-bolt lock, and 7 lock cotter pin.
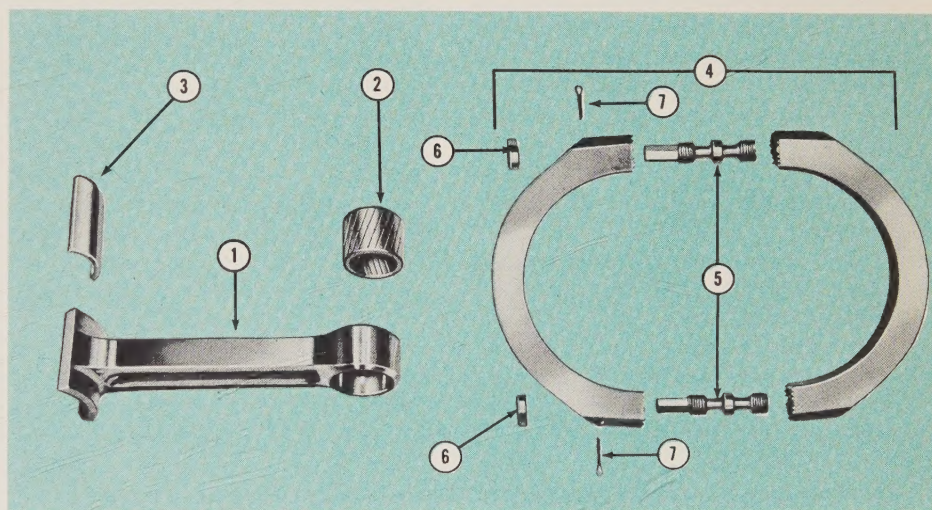
with a bronze lining and a lead-tin overlay. The bronze lining is 0.020 in. to 0.030 in. thick and the lead-tin overlay is about 0.001 in. thick.

The connecting rods, which are made of alloy steel and machined all over, are of the slipper-type (Fig. 13). The segmental bearing shell, element 3 in Fig. 13, is about 1/8 in. thick and is *dove-tailed* into the crankpin-bearing end of the rod for easy removal and replacement in service. This steel shell is lined with bronze 0.015 in. to 0.020 in. thick with a 0.001 in. thick lead-tin overlay. The wrist-pin bushing is made of steel, lined with bronze, tin-lead plated, and pressed into the eye of the connecting rod. Two retainer rings hold the four connecting

Fig. 14—Disassembled component parts of the piston which is made of a high-grade alloy steel. The trunnion, which carries the wrist pin, is bolted to the inside of the piston.

Fig. 15—Piston-cooling arrangement. The arrows indicate the path of the cooling oil as it leaves the stationary spray nozzles, enters the piston trunnion to the underside of the piston combustion chamber, and then returns to the crankcase.

Fig. 16—Cutaway view of the two-piece water-jacketed-type engine cylinder. Identified elements are 1 steel liner, 2 air-intake ports, 3 lockwire, 4 synthetic-rubber seal rings, 5 water-deflection baffle plate, 6 water-inlet port, 7 brass water-flow ferrules, 8 water jacket, and 9 seal-ring grooves.

of wooden models, the best cross section and contours to suit design and ease of manufacture were established.

Fig. 14 shows the disassembled component parts of the piston. The piston is made of a high-grade alloy steel. The crown is a steel forging flash-welded to the skirt under the ring belt. A trunnion which carries the wrist pin is bolted to the inside of the piston. There are four compression rings and two single-hook oil-scraper rings which are loaded by inner expanders to control the oil consumption.

The piston-cooling arrangement is shown in Fig. 15. Two stationary, side-by-side spray nozzles with 0.098 in. diameter orifices project a constant stream of oil through two taper holes in the piston trunnion to the underside of the piston combustion chamber. The oil is returned to the crankcase through identical holes on the opposite, bottom side of the trunnion. Also, up-the-rod cooling of the center of the piston is provided during about 60° of crank angle.

The cylinder is of the two-piece, water-jacketed type, consisting of a mild steel liner and jacket (Fig. 16). The bore of the liner is chrome-plated for greater life. The water jacket 8 is sealed at the top and bottom with synthetic-rubber seal 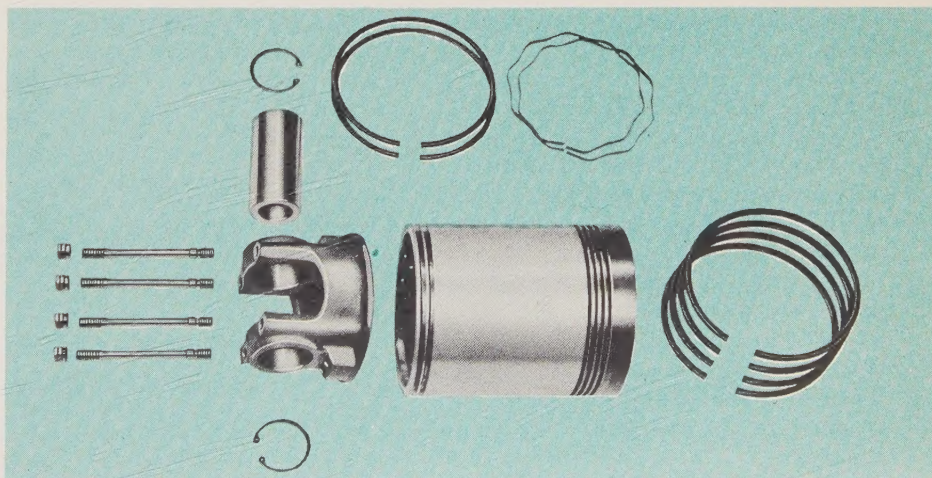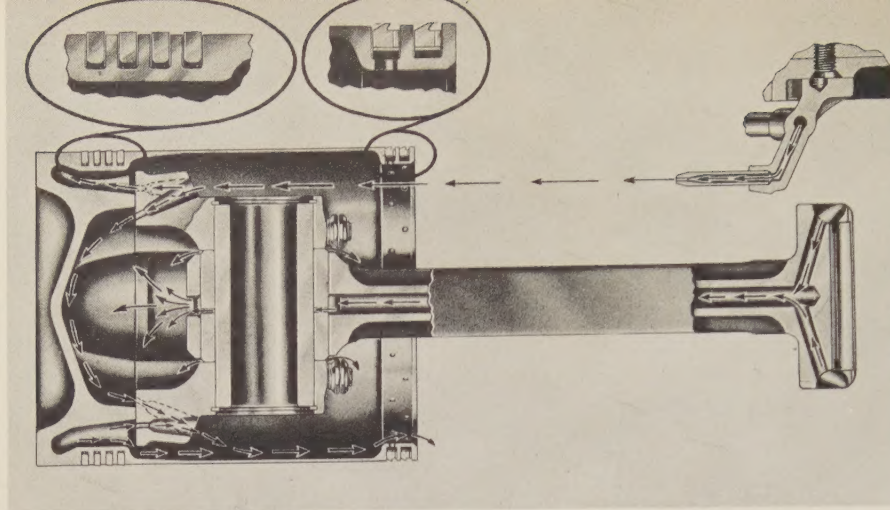rings 4. The water inlet is at 6. A baffle plate 5 is provided at the inlet to deflect the water down and circumferentially for efficient cooling. From the jacket the water flows through brass ferrules 7 into the cylinder head. No water cooling is provided below the air-intake ports 2. The cylinder is seated in the barrel forging and outer-deck forging of the crankcase and is held in the crank-

case by the cylinder head. Synthetic-rubber seal rings inserted at 9 seal the air box from the inner crankcase. There are 30 air-intake ports in the cylinder liner which admit air from the air box. The joint between the cylinder and the cylinder head is made gas-tight by a bronze gasket ring.

During the manufacture of the experimental pilot engine, various production problems had to be solved such as the automatic welding of the crankcase which was a unique undertaking in the field of welded structures. The welding processes, which were developed for the crankcase, proved to be entirely successful.

In the case of the main-bearing carriers, the fitting of the three component sections, the proper tightening of the section bolts, and the fitting in the crankcase with proper clearances between case and carrier to prevent fretting and yet provide ease of assembly had to be carefully worked out.

The proper manufacture of the dove-tailed connecting-rod bearing shell presented another problem. The shell had to be a crush fit, but not too tight to facilitate removal.

Pilot-Model Testing

After completion, the engine pilot model was installed in a special test cell with field-service conditions simulated as nearly as possible. A test program was then established to determine optimum injector and exhaust timing in order to obtain best fuel consumption and a clear exhaust. A number of cylinder liners with various designs of intake ports were tested to obtain optimum scavenging conditions.
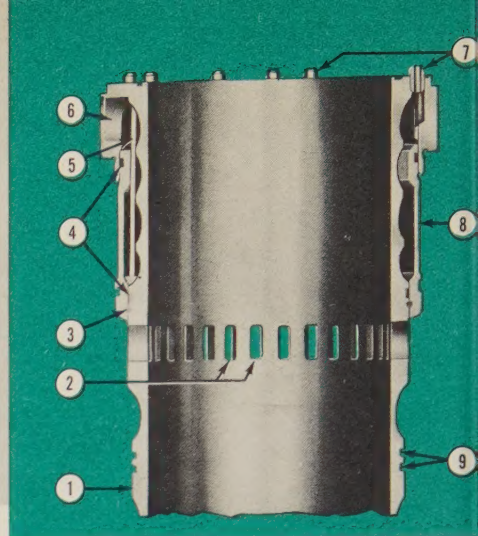
Several designs of lubrication-oil control rings were tried to obtain satisfactory lubricating-oil consumption without danger of scoring the liner. Torsiograph tests proved that torsional critical speeds occurred at positions previously calculated. Linear-vibration tests were conducted to study the requirements for suitable engine mounts.

The engine was then subjected to cyclic operation with constantly varying loads and speeds under the most severe and unusual conditions, such as high exhaust pressures and with the engine in various inclined positions.

Another time-consuming process was the study of the action of various lubricating oils on bearing materials and the optimizing of their relationship.

Experimental tests of this sort are time-consuming; in fact, they never end since further improvement is always possible.

Summary

The design of this engine and the approach is to be considered an unusual development with a penetration into many phases of engine design which were unique. Full utilization of every known means of experimental engineering with the facilities of a modern electronic instrumentation laboratory were essential. The final design was made possible, above all, by an engineering force whose members had to possess ingenuity, vision, imagination, perseverance, courage, and the required specialized knowledge.

Helium-Nitrogen Shielding Gas Improves Welding of Low-Carbon Steel

By EUGENE L. TURNER and
WALTER F. HELLER

Buick-Oldsmobile-Pontiac

Assembly Division

Inert-gas metal-arc welding, a development of the 1920's, received its biggest impetus during World War II. Welcomed were its advantages of corrosion resistance, no flux cleaning, high welding speed, and reduced grinding requirements. Shielding gases, in most applications, have been either helium or argon, or a mixture of the two. Other gas mixtures which were attempted seemed less suitable. Now, experience at the Kansas City plant of the Buick-Oldsmobile-Pontiac Assembly Division indicates that a helium-nitrogen mixture may add new benefits in welding low-carbon steel while retaining all of the other benefits of inert-gas metal-arc welding.

More heat, less

current welds

low-carbon steel

NITROGEN mixed with helium to form the shielding gas for inert-gas metal-arc welding gives promise of significant cost reduction and improved weld quality in the welding of low-carbon steel. This was revealed at the Buick-Oldsmobile-Pontiac Assembly Division, Kansas City, Kansas, after comparison tests were made using the helium-nitrogen mixture and the widely used helium-argon mixture.

A 50 per cent reduction in amperage and lower cost per cubic foot of the helium-nitrogen gas mixture were among the improved properties observed for the new method.

Nitrogen has an important effect upon the electrical behavior of the welding arc. In any welding arc, the temperature of the positive column is determined by several conditions, namely, (a) the arc current, (b) the heat conduction and convection properties of the arc atmosphere and environment, and (c) the effective ionization potential of the arc gases.

A change in any one of these factors produces corresponding changes in the voltage gradient and arc-column diameter, such that the heat produced in the arc is equal to that transferred to the arc environment. Increasing the heat-transfer ability of the arc atmosphere reduces the arc-column diameter and increases the temperature and the voltage gradient in the arc column.

Nitrogen is a diatomic gas. Like most diatomic gases, when present in the arc atmosphere it is completely disassociated at arc temperature. The recombination of single atoms to form diatomic gas molecules results in the liberation of heat to the arc atmosphere, increasing its heat-transfer ability. Thus, the arc temperature and voltage gradient are increased while the current is decreased.

Comparison Tests

In the tests that were made at the Kansas City plant of the B.O.P. Assembly Division, identical joints were welded using the two different gas mixtures. Butt welds and fillet welds in the form of T-joints were made, using 0.125-in. low-carbon steel. The steel was in the "as sheared" form with no attempt made to remove burrs, rust, or oil. Preparation of the joint was limited to tack welds at the ends of the fillet-weld joints and at one end of the butt-weld joints.

Better Weld Quality

The tests of the helium-nitrogen and helium-argon weld specimens compared

BEND-TEST RESULTS

Helium-Nitrogen — Butt Welds			
Sample No.	1	2	3
Fracture	No fracture in weld	No fracture in weld	No fracture in weld
Helium-Argon — Butt Welds			
Sample No.	1	2	3
Fracture	No fracture in weld	1/16-in. fracture in weld	Fracture across weld

Table I—Results of the bend test on three sample welds made with each gas mixture. Specimens, 1 in. by 6 in., were cut from 6 in. by 6 in., 0.125-in. low-carbon steel plates butt welded together.
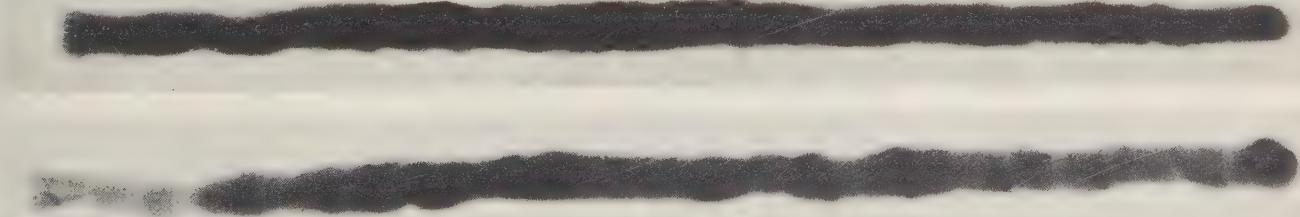
Fig. 1—(Top) X-ray of weld made using the helium-nitrogen gas mixture shows a freedom from gas pockets, heat cracks, and other welding defects. (Bottom) X-ray of weld made using the helium-argon mixture indicates the presence of heat cracks at the beginning and end of welds with scattered gas pockets throughout the welds.
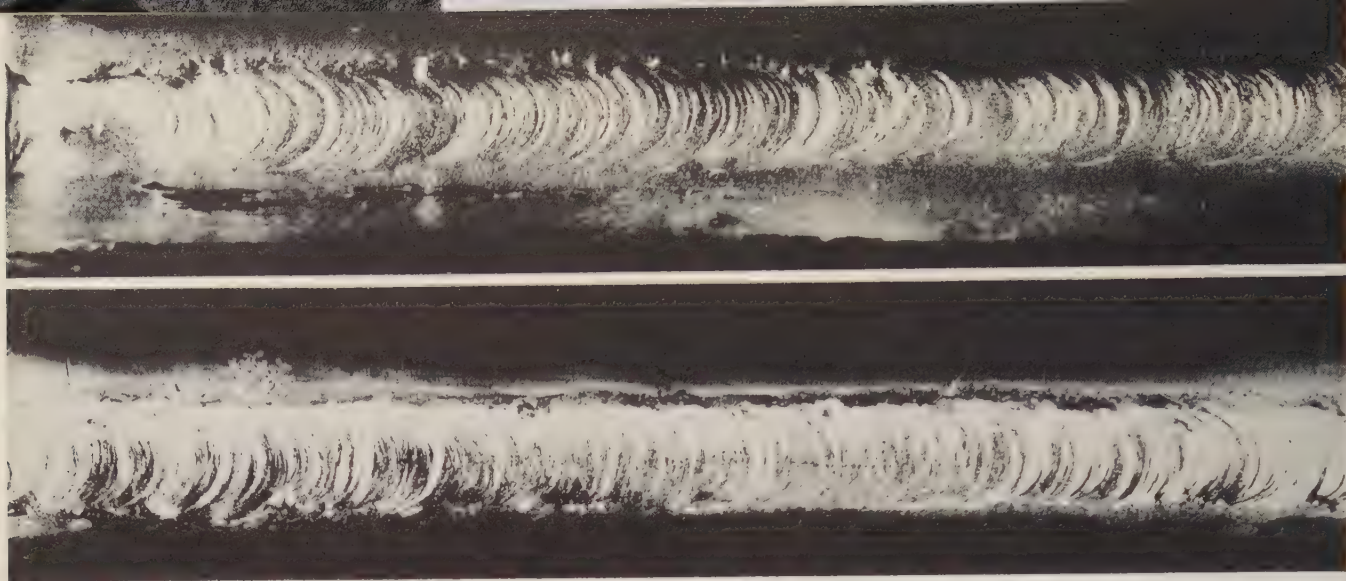
Fig. 2—Comparison of fillet welds indicates no apparent difference in the microconstituents of the weld produced by the two gas mixtures. (Top) A weld made using the helium-nitrogen mixture. (Bottom) Weld made with a helium-argon mixture.

the following characteristics:

- Ductility
- Microstructure
- Hardness
- Tensile strength
- Fatigue.

The bend test was made on weld samples 1 in. by 6 in. cut from 6-in. by 6-in. plates which had been welded together from one side only. The test samples from both types of mixed gas were polished and bent in the weld 180° over a diameter twice the thickness of the welded plates (0.250 in.). The side of the weld on which the metal was deposited was on the outside of the bend on this test. Table I shows that two of the samples of helium-argon butt welds fractured. On the other hand, none of the samples of helium-nitrogen butt welds showed fractures, indicating that helium-nitrogen welds have superior ductility over helium-argon welds.

The presence of heat cracks and gas pockets in the weld was studied by radiographs. Both butt-weld and fillet-weld test plates were examined before being cut for further testing. X-rays of these welds are shown in Fig. 1. The weld made in the helium-nitrogen mixture indicates a freedom from gas pockets, heat cracks, or other welding defects (Fig. 1, top). The helium-argon test welds contained heat cracks at the beginning and end of welds with scattered gas pockets throughout the welds (Fig. 1, bottom).

The possibility that the new gas mixture might result in the formation of undesirable nitrides, which tend to cause embrittlement of the weld, also was a consideration in these tests. Examination of the microstructure of cross sections from each type of weld was made using either nital or pical etchants. No significant difference in the microstructure of the two types of welds could be found. Both were typical of low-carbon steel welds.

Another test using ferricyanide etching solution to differentiate between nitrides and carbides was attempted. This produced no results since etching of the surface did not occur.

Hardness readings taken on the test-weld cross sections were 81 Rockwell-B Rc-B for the helium-nitrogen weld and 78 Rc-B for the helium-argon weld. This was not considered to be a significant difference in hardness which could be attributable to the gases used.

Metallographic examinations indicated no apparent difference in the microconstituents of the welds produced by the two gas mixtures. Fillet welds using helium-nitrogen gas and helium-argon gas are shown in Fig. 2.

The fatigue properties were tested with weld samples 3 in. by 10 in. cut from the plates. Compression stresses and tension stresses were equal. The results showed a scatter band of values and no conclusive fatigue-test summary could be reached to show any superior fatigue property of one weld over the other.

Tensile-strength results are shown in Table II, using samples 1 in. by 10 in. Neither gas mixture appears to have any superiority over the other in affecting the tensile strength.

Cost Reduction

In addition to the above physical properties of the welds made with the

TENSILE-TEST RESULTS

Helium-Argon Specimen No.	1	2	3
Area	0.0928	0.0927	0.0927
Ultimate Load	4,250	4,250	4,165
Ultimate Stress (psi)	45,750	45,800	44,990
Elongation (per cent in 2 in.)	30.5	30.5	30.5
Location of Fracture	Out of Weld	Out of Weld	Out of Weld
Helium-Nitrogen Specimen No.	1	2	3
Area	0.0927	0.0927	0.0929
Ultimate Load	4,250	4,250	4,115
Ultimate Stress (psi)	45,510	45,510	45,340
Elongation (per cent in 2 in.)	30.5	30.5	30.5
Location of Fracture	Out of Weld	Out of Weld	Out of Weld

Table II—Summary of tensile-test data comparing three weld specimens, 1 in. by 10 in., 0.125-in. low-carbon steel.

Fig. 3—Sectional view of fillet weld showing the penetration obtained using argon gas as the shielding medium.
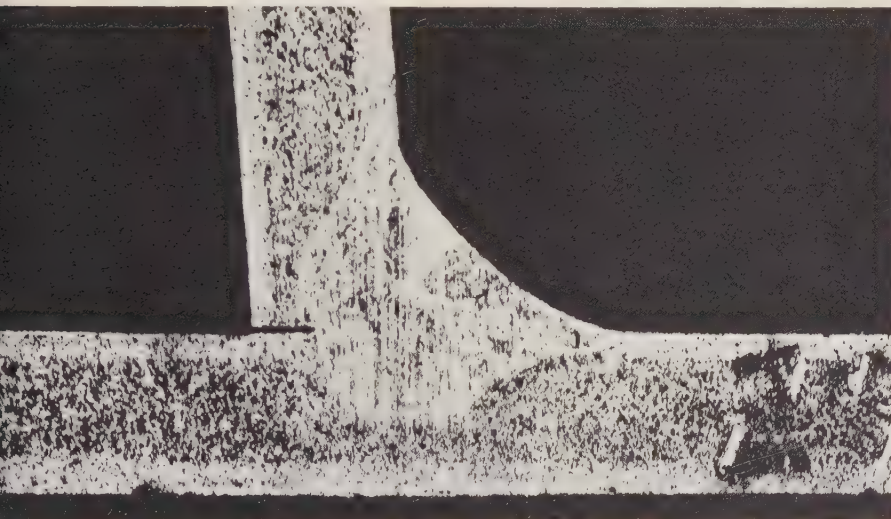
Fig. 4—Similar penetration is obtained in a fillet weld using a helium-argon mixture as the shielding gas.
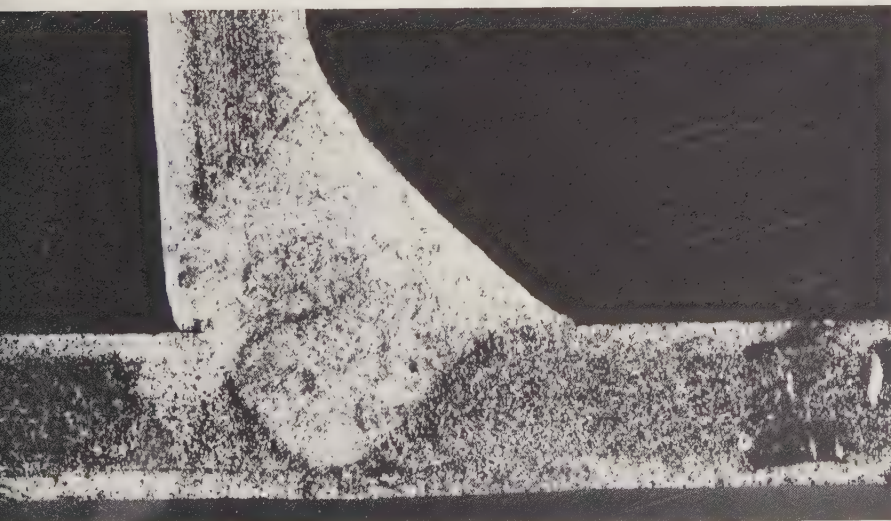
Fig. 5—In this fillet weld, a slight improvement was noted when the helium-nitrogen gas mixture was used.

helium-nitrogen shielding-gas mixture, four important cost savings in the process were determined from the test.

The welding current reduction was 50 per cent. Penetration comparisons using the three shielding gases are shown in Figs. 3, 4, and 5.

Lowered gas consumption also was a result since a decrease in amperage in inert-gas welding brings a decrease in gas flow. The ratio of the cost per cubic foot probably varies, depending on the location and the availability of supplies.

A 1/16-in. filler wire was used with the helium-nitrogen gas for single-pass fillet or butt welds. This is a saving in filler material over the 5/64-in. or 3/32-in. filler wire normally used with helium-argon gas.

Greater welding speed and a smaller, cleaner bead were other results obtained. The faster, forehand welding technique was used for fillet welds making possible larger concave beads. Undercutting was eliminated because the helium-nitrogen welds permitted good weld-metal wetting action, resulting in a minimum restriction in the flow of weld metal up the sides of the fillet legs.

The backhand technique restricts the welder's view as to direction of travel since the unwelded portion of the joint lies directly in front of the arc.

The smaller bead in a production application means that less grinding is required. This, in turn, represents a saving, for very often the grinding costs of a joined part exceed the welding costs by a considerable amount.

Conclusion

The helium-nitrogen gas mixture in inert-gas metal-arc welding appears to offer new ways to cut overall manufacturing costs and to gain an improvement in product quality.

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Design Considerations Applying to Specification of Surface Finish for Machined Parts

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Length, width, and height—these three dimensions the designer specifies on the engineering drawing. Always he considers and often he must specify a fourth dimension which may be as important as any of these three. This fourth dimension is surface finish.* Until recent years there was no widely accepted standard for specifying finish. The engineer and designer were able to tell the machinist only in general terms what would constitute acceptable roughness. Now, by industry-wide use of a standard method, the designer is able to communicate through numerical symbolism on engineering drawings precisely the limits within which the machinist must work. This system was made feasible after the introduction of adequate instrumentation for production investigation of smoothness. While the new system gives the designer greater control than heretofore, it also places on him a greater responsibility for knowing what finish to specify—or when to omit that specification. Generally, the ideal finish is the roughest which will enable the part to retain all its required characteristics and to fulfill its intended function.

Since the part is not smooth, how rough can it be and do its job?

THE development of the automobile, the airplane, and other high-speed machines has resulted in heavier loadings and increased speeds of moving parts. To withstand these more severe operating conditions with a minimum of friction and wear, a particular surface finish is often essential. This increasing importance of machined surface finish has brought about a need for accurate control of surface quality.

Until recently, it has not been possible for the engineer and the designer to write suitable specifications for surface finish or for the shop to work to definite standards. The finish of a surface was usually indicated by various familiar grind or finish marks, such as g, f, ff, or f_g. These simple markings were adequate in cases in which all the parts of a machine were manufactured and assembled in one plant, because there was close communication between the engineer, the designer, and the machinist. These markings are not adequate, however, in the modern mass-production system. Machines are rarely designed and manufactured in their entirety in one plant. The parts are often designed in one location, manufactured in one or more other places and, frequently, assembled in a third place. Under these decentralized mass-production conditions confusion may arise over the meanings of surface-finish symbols such as those in use heretofore. Consequently, there may be difficulty in getting people in various shops to work to the same degree of surface finish.

A conventional system is now available whereby surface finish can be accurately designated and measured, the basis of the system being the measurement of the average heights of the surface irregularities. This system has been approved and published as an American Standard and is now used throughout industry. It describes the surface finish by specifying the *average roughness*, the *waviness*, and the *lay* of the surface.

Standard Specifications for Surface Quality

All machined surfaces are composed of thousands of irregularities of various lengths, widths, and heights, and there are even smaller irregularities on the flanks of the major ones. The surface irregularities of some typical machined surfaces are shown in Fig. 1. These profile views suggest that a surface has depth as well as length and width and, thus, is actually a three-dimensional boundary between the material and its environment. The magnification of these profiles is approximately 10 times greater in the vertical direction than in the horizontal direction so that the depths of the irregularities are greatly exaggerated in relation to their width.

The geometry of a surface is exceedingly complex and to describe it completely would require an infinite number of parameters. It has been found adequate for production control, however, to describe the surface roughness by means
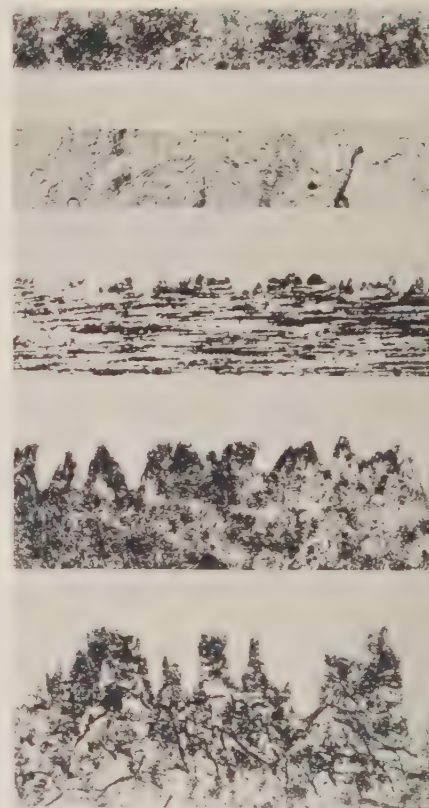
Fig. 1—Microphotographs of typical taper sections of machined surfaces. (Top to Bottom) Average roughness, in microinches, of the surfaces are: wrist pin, 4; connecting-rod bushing, 8; valve stem, 24; cylinder bore, 60; and brake drum, 110. The angle of cut used in preparing specimens for microphotography was 5° 45′. Since the width-to-height ratio of the microphotograph depends on the sine of the angle of taper (in this case 0.1), the width-to-height ratio of the above sections is 1 to 10 and the irregularities shown, therefore, are actually ten times as high. In preparing a specimen for examination, it is first nickel-plated, then taper-cut, and finally polished and etched for viewing and photography.

*For a glossary of surface-finish terms, see p. 17.

a = 4	a² = 16
b = 19	b² = 361
c = 23	c² = 529
d = 16	d² = 256
e = 31	e² = 961
f = 20	f² = 400
g = 27	g² = 729
h = 20	h² = 400
i = 31	i² = 961
j = 13	j² = 169
k = 23	k² = 529
l = 15	l² = 225
m = 6	m² = 36
Totals 248	5572
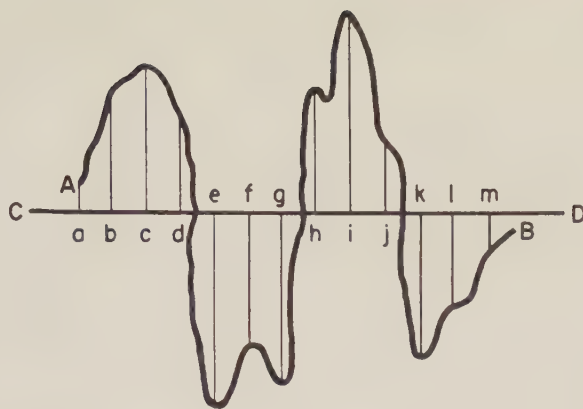
$$\text{Arithmetical average} = \frac{248}{13} = 19.1 \text{ microinches}$$

$$\text{Root mean square average} = \sqrt{\frac{5572}{13}} = 20.7 \text{ microinches RMS}$$

Fig. 2—Relationship of arithmetic-average and RMS values for determining surface roughness. As an approximation, the RMS value is considered to be the arithmetic average plus 10 per cent. Most standards specify use of arithmetic-average values.

of a single number related to the average height of the irregularities measured from the mean surface. This number also gives excellent correlation with service performance. Consequently, this method of specifying a surface has been universally adopted.

With the advent of this system, the first attempts were made to standardize the terms and methods of designating surface quality. Standards for surface finish have been set up by a number of companies and standardizing groups, including the Society of Automotive Engineers, the National Aircraft Standards Committee, and Department of Defense agencies.
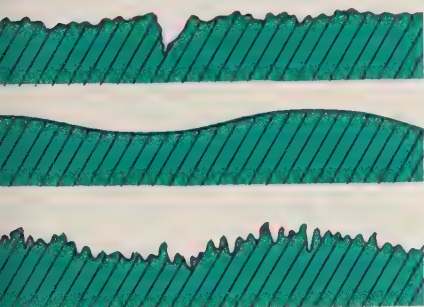
Fig. 3—Trace of an actual surface profile illustrating roughness (top), exaggerated profile of wavy surface (center), and profile of roughness superimposed on waviness (bottom).

† In this paper, the GENERAL MOTORS ENGINEERING JOURNAL deviates from the American Standards Association's abbreviations for engineering and scientific terms and follows the conventional practice of surface-finish literature by capitalizing the terms AA (arithmetic average) and RMS (root mean square).

The American Standards Association coordinated an effort at standardization among many segments of industry, technical societies, and engineering education. The preparation of this standard was sponsored by the American Society of Mechanical Engineers and the S.A.E. After numerous meetings and preliminary publications, American Standard B46.1, entitled "Surface Roughness, Waviness and Lay," was approved and published in 1947. It was intended to bring the several standards into agreement with one another and to promote national standardization. This standard defines finish terminology, establishes definite classifications for roughness, waviness, and lay, and provides a set of symbols for specifying surface finish on drawings.

At the present time, the 1947 publication is being revised and brought up-to-date. The latest proposed revision goes into greater detail in many more areas of surface specification than did the 1947 publication. The proposed standard also provides specifications for precision-reference specimens, roughness-comparison specimens, and established requirements for tracer-type instruments. The proposed standard, in common with the 1947 version which it is to supersede, is concerned with the geometric irregularities of the surface and does not treat other surface qualities such as luster, appearance, color, corrosion, wear resistance, hardness, microstructure, and like qualities—any of which may be the

governing consideration in a specific application.

Evaluation of Roughness Height

Two different parameters may be used to evaluate the average roughness of a surface. They are the *arithmetic-average deviation (AA)*† of the surface from the mean surface and the *RMS deviation* from the mean surface. These two types of averages are closely related; however, for most machined surfaces the RMS average is approximately 10 per cent greater than the arithmetic average.

The RMS average has been widely used in the past. At the present time, however, the arithmetic average is preferred and is specified as the standard rating for surface roughness in practically all of the standards on surface finish. Besides the latest proposed revision to the American Standard B46.1, other standards on surface finish include

- *British Standard 1134:1950,* "The Assessment of Surface Texture-Centre-Line-Average Height Method," published by the British Standards Institution
- *NAS-30,* "Surface Roughness Designation," published by the Aircraft Industries Association of America in 1948
- "S.A.E. Surface Finish Standard," part of the *1953 S.A.E. Handbook.*

The meanings of these terms, arithmetic average and RMS, are illustrated in Fig. 2. The curve **AB** represents a typical surface profile and the straight line **CD** represents the mean surface. The mean surface is indicated by a straight line on the profile located in such a way that the algebraic sum of the areas included between the mean surface and the true surface is zero; areas above the mean surface are considered positive and those below negative.

The RMS average may be calculated from the profile by measuring the vertical distances from the mean line to the profile at a number of equally spaced intervals along the mean line, squaring the measurements, summing the squares, dividing by the number of measurements, and taking the square root of this quotient.

The arithmetic average may be obtained by measuring the vertical distances from the mean line to the profile at a number of equally spaced points, summing these measurements, and dividing by the number of measurements. The arithmetic average is 1/4 to 1/5 the total

	ROUGHNESS-WIDTH CUTOFF	AVERAGE ROUGHNESS READING
TRUE SURFACE PROFILE		
	.030	130–150
	.010	68–80
	.003	35–40

|←———— .030 ————→|

Fig. 4—True surface profile and profile as determined by three separate roughness-width-cutoff figures, in inches. Even as the cutoff figure is reduced, indication of short wave-length irregularities are preserved. Decreasing the cutoff figure results only in eliminating indications of long wave-length irregularities, which then fall under the *waviness* classification. A preferred value is 0.030 in.

peak-to-valley height for machined surfaces. For the finer finishes, however, this ratio may be as small as 1/10.

One of the reasons that the arithmetic average is preferred is that the production and inspection personnel understand it more readily. In addition, RMS average cannot be readily measured. The instruments which are professed to read RMS average actually measure something nearly proportional to arithmetic average. A third reason for preferring the arithmetic average is that eventual international standardization is brought closer because the British now use the arithmetic average.

Fortunately, switching from the RMS system to the arithmetic-average system does not necessitate changing the roughness values which are specified in RMS units on existing drawings. As mentioned previously, the arithmetic-average roughness value for most machined surfaces is about 10 per cent less than the RMS value. Thus, the numerical difference between the roughness values in the two systems is small enough and in the right direction so that all surfaces which are smooth enough to pass an RMS inspection will also pass an arithmetic-average inspection.

Roughness-Width Cutoff

Fig. 3 illustrates conditions of roughness (top) waviness (center) and roughness superimposed on waviness (bottom). In Fig. 3 the closely spaced irregularities of short wave lengths constitute roughness, while the widely spaced irregularities of longer wave lengths constitute waviness. For this idealized surface it is easy to differentiate between the two. For actual machined surfaces, however, it is often difficult to distinguish between roughness and waviness because there are irregularities of all wave lengths present and the short wave lengths flow smoothly into the long wave lengths. The *roughness-width cutoff*, therefore, may be used to differentiate between the two.

The roughness-width cutoff is the maximum width in inches of surface irregularities to be included in the

Fig. 5—The new surface symbol may be drawn with its point on a line indicating the surface, on a witness line, or a leader line pointing to the surface. Widening the short line is optional.

measurement of average roughness. The cutoff determines the distance over which the surface irregularities are to be averaged to obtain the roughness value. Irregularities having spacing greater than the roughness-width cutoff are
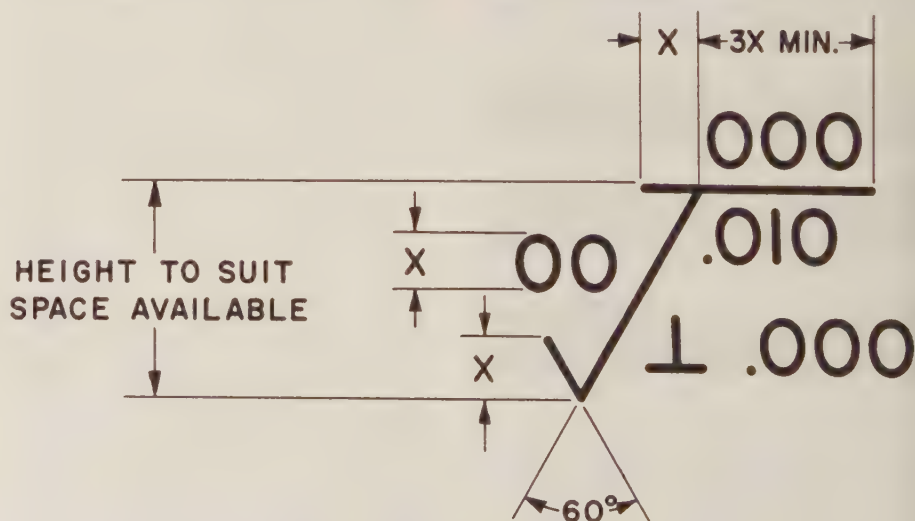
Fig. 6—The new surface symbol provides easily understood numerical specification for roughness to replace the older system which used notes to tell the manufacturing department how to do the job. The zero is omitted before the decimal point for maximum space utilization.

called *waviness* and are not included in the measurement of roughness.

Fig. 4 shows the true profile of a machined surface and its interpretation by an instrument with roughness-width cutoff settings of 0.030 in., 0.010 in., and 0.003 in. Each profile shows only those irregularities that are sufficiently close together to be measured as roughness for the particular roughness-width cutoff. The very short irregularities occur in all the profiles. The measuring instrument, however, is less sensitive to irregularities having a wave length greater than the roughness-width cutoff. For each profile all irregularities have been attenuated that have a wave length longer than the cutoff. The profile for the 0.030 in. cutoff is little different from the true profile because in the surface there are no irregularities with a wave length longer than 0.030 in. In the profile for the 0.010 in. cutoff, the coarser irregularities—those with a wave length longer than 0.010 in.—have been reduced in amplitude and the finer irregularities are relatively unchanged. The profile for the 0.003 in. cutoff includes only the irregularities with a wave length less than 0.003 inch.

For this particular surface profile, the effect of reducing the roughness-width cutoff—thereby excluding some of the irregularities—has been to decrease the average roughness value, which is indicated by the values given. On the other hand, if all of the surface irregularities were as fine as those of the bottom profile, the roughness value would be the same for all three roughness-width cutoff settings.

For certain applications, adequate control of surface roughness may require the use of a particular roughness-width cutoff. For example, a large cutoff value is used where the contact area between two mating surfaces is important. Conversely, in cases where parts may be subject to fatigue failure, only the surface scratches which serve as stress raisers are important and a small cutoff gives a more significant value. In selecting a roughness-width cutoff, care must be taken to choose one that measures important irregularities.

The standard roughness-width cutoff values are 0.003 in., 0.010 in., 0.030 in., 0.100 in., 0.300 in., and 1.000 in. Experience has shown that the 0.030 in. value is preferred for most applications, and this value is used unless there is a definite reason for specifying a different one.
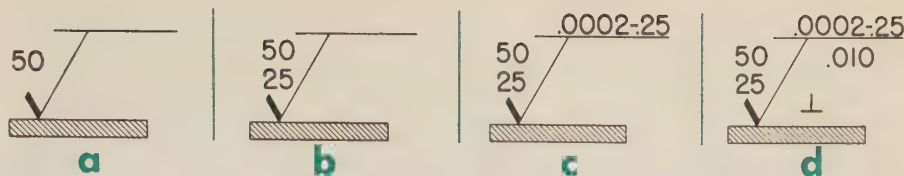
Fig. 7—Typical surface-finish symbolism. One number, as in 7a, denotes average roughness. Two numbers, as in 7b, show permissible maximum and minimum roughness values. Fig. 7c has waviness-height rating in inches added. In Fig. 7d, the lay symbol (perpendicular to edge of surface indicated) and roughness-width-cutoff value have been added.

Surface-Finish Symbols

The designer specifies surface finish on his drawings by means of the proper symbols for surface roughness, waviness, and lay. Generally these symbols replace notes or symbols calling for a specific method of machining.

A surface can be more accurately described and duplicated by the proper use of surface symbols than by specifying the manufacturing process, an operation which may be subjected to many different interpretations. The designer specifies the surface finish, and unless experience indicates that only one processing method will give adequate performance, the decision on the method of machining is left to the manufacturing department which has full knowledge as to which means of producing the parts will best fit its equipment, facilities, and schedules.

The symbol specified by the latest proposed revision to the American Standard B46.1 for designating surface finish is the check mark and extension shown in Fig. 5. This mark is put on the drawing wherever control of surface finish is required. The point of the symbol may be on a line indicating the surface, on a witness line, or on a leader line pointing to the surface. The short leg may be wider if desired. The proportions for the surface-finish symbol are shown in Fig. 6.

The method of designating roughness, waviness, roughness-width cutoff, and lay in conjunction with the surface-finish symbol is shown in Fig. 7. Only those ratings that are necessary to specify the required surface should be shown on the symbol.

The average roughness value is placed adjacent to the left side of the long leg as shown in Fig. 7a. The use of one number indicates the maximum permissible roughness value; any lesser value is acceptable. When two numbers are used, as in Fig. 7b, they indicate the permissible maximum and minimum values of roughness.

The waviness-height rating, when required, is placed above the horizontal extension line (Fig. 7c). This rating represents the maximum peak-to-valley height of waves in inches and any lesser value is acceptable.

Where required, the waviness-width rating, not shown in Fig 7, is placed immediately to the right of the waviness-height rating. The number represents the maximum value in inches and any lesser value is acceptable. A percentage contact-area value may be used as an alternate to waviness-width rating.

The roughness-width-cutoff value is placed to the right of the long leg and directly below the horizontal extension line (Fig. 7d). The value of the roughness-width cutoff is not designated unless a value other than the standard 0.030 in. is required.

Lay designation is indicated by the lay symbol placed to the right of the long leg (Fig. 7d). The symbol is used only if considered essential. The standard symbols for designating lay are shown in Fig. 8.

Design Considerations

A comprehensive program of surface-finish control must clearly state which surfaces are critical. Where no surface finish is indicated, any machining operation that gives the required dimensional accuracy also gives an adequate surface finish. On many surfaces, finish control is not required because service life is not affected by the surface finish. It is important that these surfaces receive no surface-finish designations on drawings; otherwise, the cost of the product may be unnecessarily increased and also the truly important surface-finish marks may receive less attention.

Proper control of surface finish can eliminate many rejects and failures and, consequently, can result in better products at lower costs. Care must be taken to avoid over-specification, however, because normally the cost of producing a surface becomes progressively greater as

═	Parallel to Edge of Surface Indicated	⊥	Perpendicular to Edge of Surface Indicated	X	Angular in Both Directions

C	Approximately Circular Relative to Center	M	Multi-Directional	R	Approximately Radial Relative to Center
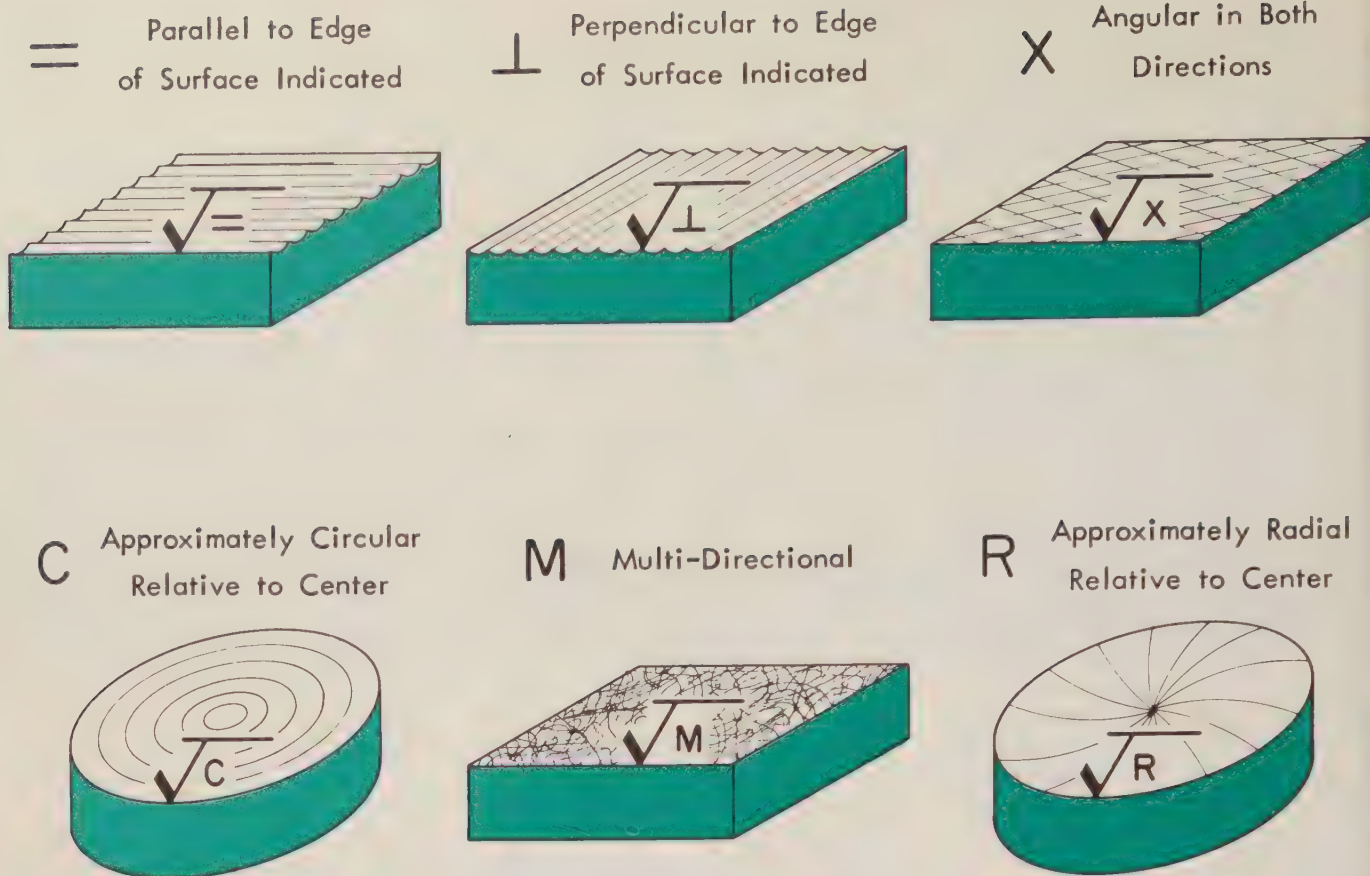
Fig. 8—Lay of the surface irregularities may be parallel or perpendicular to the edge of the surface or angular in both directions; they may be approximately circular relative to the center or approximately radial. Irregularities also may lay multi-directionally. There is a standard symbol for designating each kind.

the permissible roughness and waviness become less. Generally, other considerations being accounted for, the *ideal finish* is the roughest one which will do the job.

The engineer has the responsibility in each case of selecting the surface finish that will give maximum performance and service life at the lowest possible cost. He bases his decision on past experience with similar parts, on engineering tests, or on field-service data. His choice will be influenced by such factors as size and function of the part, type of loading, existence of load reversals, speed and direction of movement, physical characteristics of materials in contact, type and amount of lubrication, contaminants, and temperature. Because of the many factors which are involved, it is often difficult to foretell accurately the proper surface finish.

The two most important reasons for surface-finish control are to reduce friction and to control wear. Wherever a film of lubricant must be maintained between two moving parts, the height of the surface irregularities must be less than the thickness of the oil film, even under

the most severe operating conditions. Bearings, journals, cylinder bores, piston pins, bushings, pad bearings, helical and worm gears, seal surfaces, and machine ways are examples for which this condition must be fulfilled.

Surface finish is important also to the wear service of certain pieces which are subjected to dry friction, such as machine tool bits, threading dies, stamping dies, rolls, clutch plates, and brake drums.

Smooth finishes are essential on certain high-precision pieces. In injectors, high-pressure cylinders, and similar mechanisms, smoothness and lack of waviness are essential in order to meet dimensional tolerances and to permit proper operation at high pressures. Smooth finishes also are necessary to the accuracy of measuring devices, such as micrometer anvils, gages, and gage blocks.

Surface finish often must be controlled for the purpose of increasing the fatigue strength of highly stressed members which are subjected to load reversals. A smooth surface eliminates the sharp irregularities which are the greatest po-

tential source of fatigue cracks.

Smoothness is often essential for eye appeal of the finished product. Surface finish is controlled for this purpose on such articles as rolls, extrusion dies, and precision casting dies. Finish control also may be essential to insure quiet operation of gears and similar parts.

In general, with compatible surfaces and complete lubrication, the smoother the surface, the better the results. However, the goal of working toward perfectly smooth surfaces is often not economical, nor is it a guarantee that the finished part will perform to the best advantage. In some applications, surfaces with a specific roughness perform better than either smoother or rougher surfaces.

For example, a specific roughness is required in order to achieve wear-in of cylinders in internal-combustion engines. As a result of imperfect geometry, running clearances, and thermal distortions these surfaces must wear-in by actual removal of metal. The surface finish must be a compromise between sufficient roughness to give proper wear-in and sufficient smoothness to give the expected service life. Too smooth a surface will

produce too slow an initial wear. In fact, the surfaces may never wear-in and improper clearances may result in local hot spots and high oil consumption. On the other hand, if the surface is too rough, the initial wear will be high, and at high rates of wear the wear particles will be large, thus acting as abrasives. The wear then will be maintained at a high rate which may continue long after wear-in should have been completed. Thus, the machine may wear-out before it wears-in.

Summary

Considerable progress has been made through cooperative efforts of technical societies, engineering educators, and industry in improving the engineering drawing as a means of communicating to manufacturing people the way in which parts are to be made. Through more specific and more easily understood terminology and symbolism it is now possible for the engineer to make more intelligent use of the qualities which accurate surface finish can give to a part. In addition, manufacturing departments are given greater latitude in deciding how to do the work. Although the engineer and the man on the machine may be thousands of miles apart geographically, they are together to the fraction of a microinch in the surface-finish task which needs to be done.

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Other related literature in this field includes the following:

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Glossary

Terms listed are those which occur in the designation of surface characteristics. Wording of the definitions are from the latest proposed revision to the *American Standard B46.1-1947*, "Surface Roughness, Waviness and Lay," except those explanatory portions enclosed within parentheses, which pertain to the text of this manuscript.

Flaws are irregularities which occur at one place or relatively infrequent intervals in the surface, e.g., a scratch, ridge, hole, peak, crack, or check. (They are not typical of the surface since they are not usually caused by the normal machining process. Acceptance or rejection of a part containing flaws is a matter of using the present recognized methods of inspection and may be based on separate engineering specifications.)

Lay is the direction of the predominant surface pattern produced by tool marks or grains of the surface ordinarily determined by the production method used.

Nominal surface is the theoretically accurate design, the shape and extent of which is usually shown and dimensioned on a drawing or description specification. (It is the surface which would result if the peaks were leveled off to fill in the valleys.)

Profile is the contour of a section perpendicular to a surface unless some other angle is specified. (Fig. 1 illustrates surface profiles.)

Roughness consists of the relatively finely spaced surface irregularities, the height, width, and direction of which establish the predominant surface pattern. Irregularities produced by cutting edges and machine tool feed may be considered roughness. The height is rated in microinches arithmetical-average deviation from the mean line, the symbol AA being used as a descriptive abbreviation. The width is rated in inches as the maximum permissible spacing between repetitive units of the surface pattern and may be specified only where roughness-width cutoff is omitted. (Arithmetical average is also known as centerline average, abbreviated *CLA*. Roughness itself does not affect the trueness of a surface. Fig. 3 (top) illustrates the departure of the actual profile from the theoretical profile as a result of roughness.)

Roughness-width cutoff is the maximum width in inches of surface irregularities to be included in the measurement of roughness height.

Surface of an object is the boundary which separates that object from another object or substance.

Surface irregularities are deviations from the nominal surface including roughness and waviness.

Waviness consists of irregularities of the nominal surface which are of greater spacing than roughness. These irregularities may result from such factors as machine or work deflections, vibration, heat treatment, or warping strains. The height is rated in inches peak-to-valley distance. The width is rated in inches as the spacing of adjacent waves. (Fig. 3 (center) illustrates the exaggerated profile of a wavy surface. Roughness may be superimposed on a wavy surface, as shown in Fig. 3 (bottom).)

A Discussion of Instrumentation for Determining Surface Roughness of Machined Parts

In the past, evaluation of the height and spacing of irregularities of machined surfaces depended entirely on the visual and tactile senses of the machinist. Engineers have long sought evaluation means which were more accurate so that in parts where a certain maximum roughness was desired the machinist or an inspector would have a means of knowing for sure when the specified smoothness was achieved. Several types of instruments have been developed during recent years and their successful application has made feasible industry-wide standardization of surface-finish designation. Among the techniques for roughness measurement in use today are optical interference methods, examination of translucent replicas and taper sections, and comparison with various types of standard finishes of known roughness characteristics. The Research Laboratories Division has made a thorough study of surface-finish measurement and has concluded that the most suitable instruments for production-line use are of the tracer-type. In these, the vertical movement of a stylus as it moves over a surface is converted into an electrical voltage, which is amplified and measured. A major contribution to surface-finish measurement is the Surfagage, a tracer-type instrument which was developed by this Division.

SINCE a machined surface consists of a succession of hills and valleys which may vary both in height and in spacing, a complete study of the surface topography would be difficult and tedious. Fortunately, present-day standards of surface-roughness specification, in conjunction with available instrumentation, permit an evaluation of surface roughness in quantitative terms.

The measurement of surface finish is accomplished by means of many different techniques. The following are the most important of these methods.

Prior to the development of instrumentation, the roughness of a surface had to be determined by using the senses of sight and touch. Surface specimens of several known roughness values produced by the various machining methods were useful for comparing with surfaces of unknown roughness. This method was, however, susceptible to human error, particularly when the surface irregularities were small. The eye—for example—tends to associate the pitch of machined marks with roughness, to associate dull surfaces with rough surfaces and shiny surfaces with smooth surfaces; however, these associations are not necessarily valid.

Microscopic Study Methods

The eye is able to resolve finer irregularities in the surface topography if supplemented by microscopes using parallel and/or oblique lighting.

Interference-Microscope Method

Very fine details can be resolved by the use of an *interference microscope*. In addition to the magnification provided in the plan view of the microscope, it utilizes the principle of optical interference to measure the height of the irregularities of the surface.

Light from a monochromatic light source is directed to both the specimen and a flat reference surface. The light

Fig. 1—Interferogram of a relatively smooth surface. The horizontal lines are interference fringes. The deviations of the interference lines which form the vertical patterns are scratches in the surface.
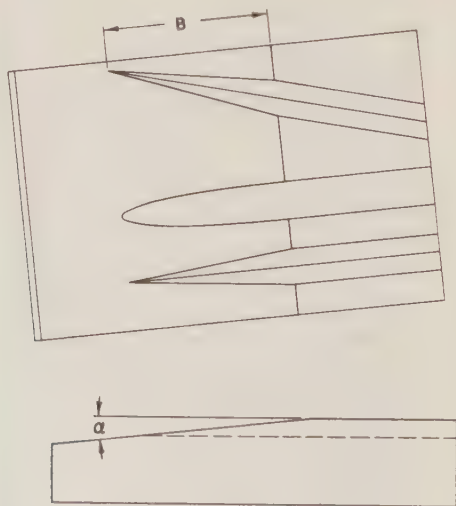
A   ACTUAL DEPTH

B   TAPER SECTION MEASUREMENT

$$\frac{B}{A} = \frac{1}{\sin \alpha} = \text{MAGNIFICATION}$$

Fig. 2—A three-dimensional view of surface irregularities shows how magnification is achieved by taper sectioning.

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Research Laboratories
Division

Now surface roughness can be

measured with precision—at

the machine

Fig. 3—The Surfagage, a portable, low-cost instrument of the tracer-type which was developed in the Research Laboratories Division.

reflected from these surfaces is recombined to form the interference fringes. A light fringe or band appears when the reflected light from the two surfaces is in phase and a dark band appears when the light waves are out of phase. The depth represented by two adjacent dark bands is one-half wave length of light or, for a green light source, approximately 10¾ microinches. Generally, the flat reference specimen is tilted slightly to form a number of light and dark bands. A perfectly flat specimen would form a number of parallel and equally spaced light and dark bands. Fig. 1 is an interferogram of a surface having a few shallow scratches. The interference bands form profiles of the surface irregularities.

This method is limited to the inspection of relatively smooth surfaces.

Translucent-Replica Method

The character of a surface also can be studied by obtaining a translucent replica. An evaluation of the surface is thus obtained, minus the often-misleading elements of color and luster existing in the original specimen. Replicas are made by moistening the surface with a solvent and then impressing a plastic tape on it. The tape is held in contact with the surface until the solvent has evaporated and is then peeled off and mounted in a suitable holder. This replica can be enlarged to show the plan-view distribution of the surface irregularities. In this way, translucent replicas can be made of surfaces which are not readily accessible for direct inspection, such as on gear teeth. They may also be used for surfaces which are not easily measured, such as spheres, for surfaces

which are specular (mirror-like), or for surfaces which are extremely soft. In some cases, this method provides a more accurate means for studying the surface than does direct visual inspection.

Taper-Sectioning Method

The taper-sectioning method can be used for an accurate determination of the contour and depth of surface irregularities. The principle of this technique is illustrated in Fig. 2. When a metal surface is taper sectioned, a supporting material must be plated over the surface in order to prevent the destruction of the surface when the taper section is ground and polished. The plated metal must have a high color contrast with the base metal of the specimen in order to provide a conspicuous line boundary between the materials. A nickel plate 0.025 in. thick usually is satisfactory for steel surfaces, the specimen being etched or heat tinted for contrast after the section is polished. A convenient angle for taper sectioning is 2° 17′, which magnifies the depth of the irregularities 25 times. About 10 microinches is the smallest peak-to-valley depth that can be resolved by this method. Two disadvantages of this method are that the test piece is destroyed in taper sectioning and the process is time-consuming.
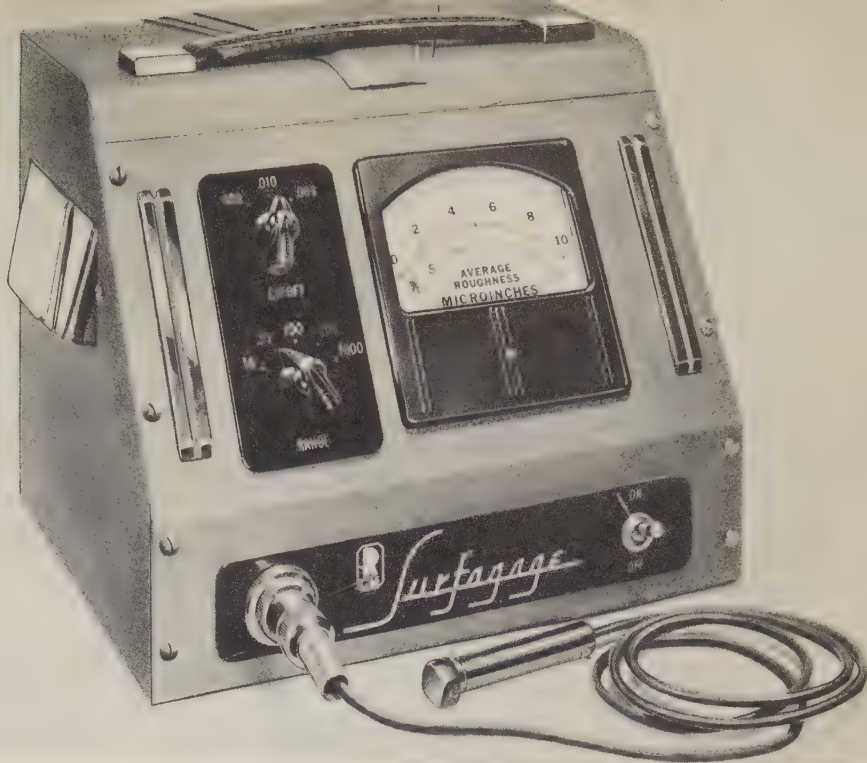
Tracer-Type Instruments

The development of the *tracer-type* instrument has made available the means of measuring surface roughness quickly and accurately. The tracer-type instrument is basically simple and relatively easy to use. It consists of a fine tracer point or *stylus* which is drawn across the surface to be measured. The vertical motion of the stylus is transformed into an electrical signal. The signal—after amplification—is measured by a meter calibrated in units of surface roughness.

There are several factors which may influence the reading given by a tracer-type instrument. On smooth surfaces the size of the stylus tip can materially affect its vertical motion. A large tip does not penetrate to the bottom of the valleys of the surface irregularities and hence the meter reading is lower than the actual roughness. A 60° to 90° cone terminating in a spherical tip of 0.0005 in. radius is satisfactory for most applications. As this radius is increased, errors in the measurement of machined and ground surfaces rapidly become greater. A smaller tip radius, on the other hand, results in extremely high load pressure which increases the wear of the stylus and, in addition, deforms the irregularities of the surface.

Another important element that can
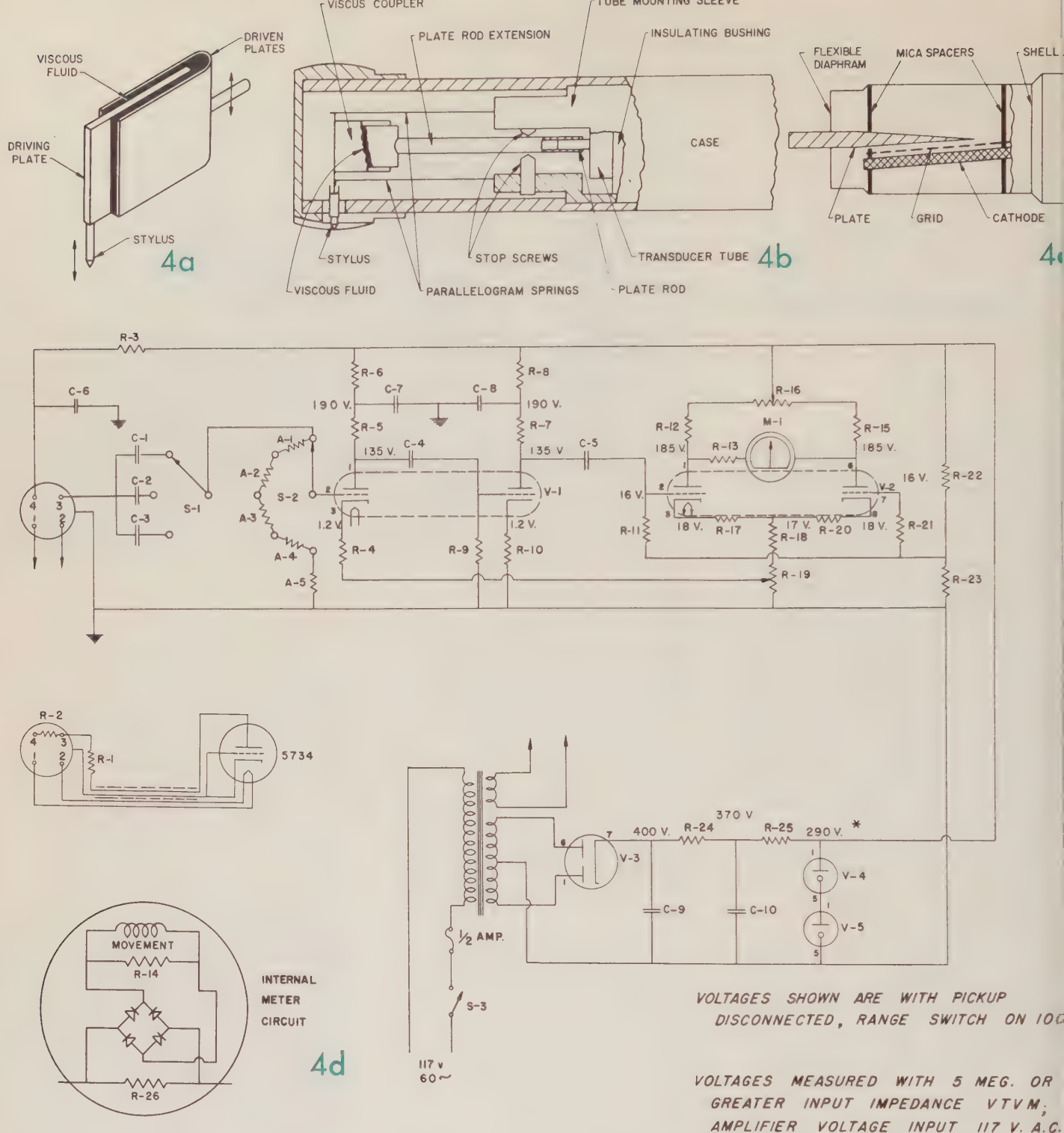
Fig. 4—A series of illustrations showing how the mechanical motion of the stylus is converted by electronics into a meter reading. The vertical motion of the stylus, Fig. 4a, is coupled to the U-shaped driven plates by means of a viscous fluid. The viscosity of the fluid is chosen so that the desired dynamic displacements of the stylus are transmitted but the static displacements caused by the pickup conforming to various curved surfaces are not transmitted. The driven plates are attached to an extension of the plate rod of the transducer tube, as shown in the cutaway sketch, Fig. 4b, and the rod and stylus assembly are allowed to move only in the vertical degree of motion by the parallelogram spring assembly. The plate rod of the transducer passes through the tube's flexible diaphragm which forms a fulcrum. The plate of the tube is attached to the plate rod as shown in Fig. 4c. Motion of the plate changes its position with respect to the grid structure and cathode of the

transducer tube. This, in turn, changes the plate current and hence changes the plate voltage of the tube. (The transducer circuit is shown as an inset at the left center.) The plate-voltage variations are introduced to the input of the amplifier through a high pass filter consisting of C-1, C-2, or C-3 and A-1 through A-5. The capacitor in the filter is dependent upon the position of switch S-1, the roughness-width-cutoff control. S-2 is the range switch whose five positions permit full-scale readings of the meter for ranges of 10 microinches to 1,000 microinches of surface irregularity. Two stages of triode amplification followed by the balanced output stage comprise the amplifier. Negative feedback is employed for constant amplification independent of powerline voltage changes. (The meter internal circuit, where the amplified signal of the stylus motion is converted to a meter deflection, is shown in the inset at the lower left.)

ffect the meter readings of a tracer-type
instrument is the maximum width of the
surface irregularities to be included in
the meter reading. The maximum width
of irregularities indicated on the meter
either is fixed by the design of the instru-
ment or—on some instruments—may be
selected by means of a *roughness-width-
cutoff control*. In the past, the importance
of roughness-width cutoff has not been
generally recognized; consequently, on
some surfaces one instrument often might
read much higher than another. Re-
cently established standards now provide
several values of roughness-width cutoffs.
At present, 0.030 in. is the most common
cutoff value and is used unless another
value is specified on the engineering
drawing.

Tracer-type instruments measure the
vertical deviation of the stylus with
respect to some reference plane. On most
instruments the reference plane is estab-
lished by skids attached to the pickup
which rides over the machined surface.
The skids must be much larger than the
irregularities of the surface, otherwise the
skids move up and down as the surface is
traversed. Any vertical motion of the
skids is superimposed on the motion of
the stylus, resulting in an error that can
either add to or subtract from the meter
reading, depending upon the geometry
of the surface and the pickup.

Two systems of designating surface-
roughness height have been used in this
country. The *root mean square* or *RMS**
system was used in the early days of sur-
face-finish measurement, but it is now
being replaced by the *arithmetic average* or
AA system. Some proponents of the RMS
system claim that the RMS value more
nearly describes the effect that the sur-
face finish has on a part's operation, but
this claim has not been substantiated.

Tracer-type instruments that are cali-
brated in RMS units do not always
measure RMS values. They indicate
about 10 per cent more than the arithme-
tic value because the meters normally
used in surface-measuring instruments
are sensitive to the arithmetic-average
current. If the wave form of the current
known, an arithmetic-average meter
an be recalibrated to indicate true RMS
values, which is done in the case of

Fig. 5—The Surfindicator, which is the production version of the Surfagage.

electric-power measurements. However,
the wave form can vary widely from sur-
face to surface and an arithmetic-average
meter cannot possibly be calibrated to
indicate RMS values for all wave forms.
An arithmetic-average meter, on the
other hand, indicates the correct AA
value for all wave forms.

The meter reading is quite steady when
extremely uniform surfaces are being
measured. On most machined surfaces,
however, the meter fluctuates over a
range of values because of the variations
in surface roughness. For these variable
surfaces a high and low reading indicat-
ing the range of roughness may be
desirable. Occasional extreme fluctua-
tions represent flaws rather than average
surface conditions and should not be used

to determine the average roughness.

The inertia and damping of the meter
reduce the speed of response of the sys-
tem, and the tracer must traverse a
sufficient distance to permit the meter to
reach its final reading. On short sampling
lengths, the pickup must be moved back
and forth several times. A skilled operator
can obtain a good roughness reading on
sampling lengths as short as the width of
a piston ring.

Several instruments of the tracer-type
are available today. No single instrument
can be considered ideal for all applica-
tions; therefore, the choice of the instru-
ment must depend on its particular
application.

One instrument in use at the time
when the Research Laboratories Division
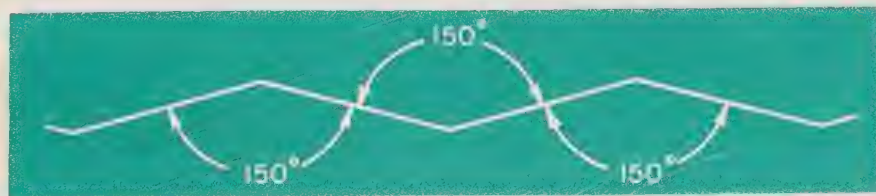
Fig. 6—Surface profile for the *Precision Reference Specimens of Surface Roughness*. The arithmetic-
average height of each specimen must be accurate to ± 1 microinch—or 3 per cent—whichever is
greater. The technique developed in the Research Laboratories Division is to generate the master
surface on gold blocks by means of a ruling machine. A diamond of the required profile is drawn across
the specially prepared gold surface to rule each groove.

began development in the surface-measurement field employed a glass standard having a surface roughness of 10 microinches to 12 microinches to check the calibration. The meter on this particular instrument indicated roughness in RMS units. A newer version of the same instrument has recently been introduced which gives the operator a choice of either RMS or AA calibration. A motor tracing unit is available for moving the pickup across the surface and the roughness-width cutoff is fixed at 0.030 in.

Another manufacturer had introduced an instrument having a motor-driven pickup, an amplifier, a meter, and a direct-inking oscillograph. The meter can be calibrated to indicate either RMS or AA surface roughness. The oscillograph gives a graphic reproduction of surface irregularities smaller than 0.010 in. width, its roughness-width cutoff. The instrument is not applicable to the full range of machined surfaces, being unable to record larger irregularities on either the meter or the oscillograph. The instrument is calibrated by means of a glass standard having a scratch of known depth inscribed on its surface.

Another model of tracer-type instrument, in extensive use abroad but only recently introduced to this country, embodies several unique features for measuring surface finish. It consists of a pick-up assembly, indicating meter, amplifier, and recorder. The pickup is moved across the surface for a fixed distance by means of a motor-driven mechanism. At the end of the stroke, when the apparatus comes to rest, the pointer of the meter does not swing back to zero like that of an ordinary meter but comes to rest at the average roughness value. Thus, the meter reading is definite, repeatable, and independent of personal manipulation. The roughness-width cutoff can be

adjusted to several values. The recorder of this instrument draws a true graphic profile of the surface being traversed by the pick-up assembly. In addition, a datum attachment is available to eliminate the use of skids and to establish a reference line for the pickup which is free from the variations emanating from the irregularities of the measured surface. This instrument was developed essentially for laboratory use and is not suitable for portable operation.

The Research Laboratories Division undertook the development of a portable instrument that would be accurate and relatively low in cost and which could be used with ease on the production line. The instrument which has resulted from these investigations is called the Surfagage (Fig. 3) and consists essentially of a pickup, which holds the tracer, and an amplifier (Fig. 4). The pickup can be used to measure the roughness of flat surfaces (cylinders) and holes larger than $\frac{7}{8}$ in. in diameter. The instrument's amplifier has a sensitivity selector which permits sensitivities with full-scale ranges of 10 microinches, 30 microinches, 100 microinches, 300 microinches, and 1,000 microinches arithmetic-average roughness. Another feature of the amplifier—

Fig. 8—This illustrates the effect of using a finite stylus radius on the roughness specimen. The reading is 0.3 per cent low for a 0.0005 in. radius stylus traversing the 125 microinch surface. Note in the lower portion of the drawing how the stylus does not penetrate to the bottom of the 20 microinch surface resulting in a reading 12 per cent low. This is the scheme used to measure the effective stylus radius.
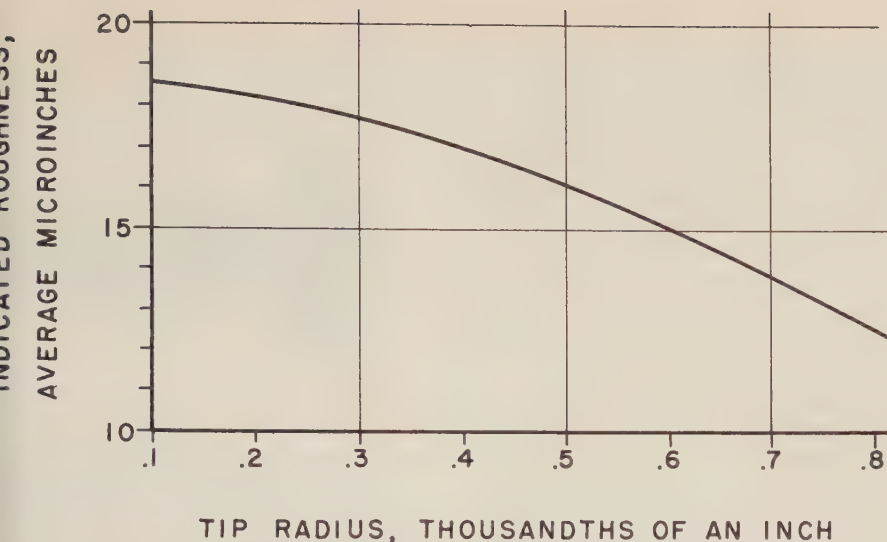
Fig. 9—Illustrated is a typical graph supplied with each set of *Precision Reference Specimens of Surface Roughness* to aid in taking stylus-tip measurements. The graph is plotted to show the actual tip radius when the instrument reading on the 20 microinch patch is known. This set is furnished with each manufactured prototype of the Surfagage, accounting for its superior accuracy in measuring surface roughness. The specimens may also be used, however, with any tracer-type instrument to check its calibration and effective stylus radius.

one that has not been generally available in this country—is the adjustable roughness-width-cutoff control which has the more commonly used 0.030 in. as well as 0.010 in. and 0.003 in. roughness-width cutoff.

The Brush Electronics Company has been licensed to manufacture this instrument for sale under the name of Surfindicator (Fig. 5). The pickup and amplifier components are the same as those used in the Surfagage. The case has been altered, however, for greater eye appeal and to provide storage space for the pickup and the calibrating roughness specimens used with the instrument. A motor tracing unit is available for this instrument. The unit has a stroke which can be reversed nearly instantaneously and, consequently, accurate roughness readings can be made on short stroke lengths. A small bore adaptor also is available to determine the roughness in holes with diameters as small as ⅛ in.

Reference Specimens Used for Calibration of Instruments

The Surfagage is calibrated by means of the *Precision Reference Specimens of Surface Roughness.* These are physical standards of surface roughness with carefully controlled surface profiles.

Before the introduction of these stand-

ards, it was necessary for the users of tracer-type instruments to depend upon the manufacturer for the calibration of their instruments, or to use a single scratch of known depth on a piece of glass for the calibration of graph-type instruments. The user could not determine if the diamond on the stylus had been worn sufficiently to cause a considerable number of errors in the measurement of surface roughness.

Wide experience has shown that the surfaces produced by conventional machining methods lack the uniformity and reproducibility required of precision roughness standards. A surface was needed that had a simple regular profile so that the RMS and AA values could be calculated and the surface would be useful for calibrating various types of roughness-measuring instruments.

The surface profile shown in Fig. 6 satisfies the above requirements and has been selected for the Precision Reference Specimens of Surface Roughness. *Proposed Surface Finish Standard B46.1* specifies that the arithmetic-average height of precision reference specimens shall be accurate to ± 1 microinch or 3 per cent—whichever is greater. The Research Laboratories Division has developed a technique for making precision reference specimens on gold blocks to the necessary degree of accuracy. The master surfaces are generated by a ruling machine similar to the type used in making diffraction gratings. A diamond having the required profile is drawn across the specially prepared gold surface to rule each groove. The roughness of each patch is controlled by changing the pitch and depth of the grooves. The interference microscope is very useful in

judging the depth of the gold master roughness specimens at the time they are being ruled.

After measuring the masters to be sure they conform to the necessary requirements of precision, nickel copies are made by an electroplating process. Additional copies are then made from those copies until the specimens which the user ultimately obtains represent the sixth generation of the originals.

At the present time, these roughness specimens are available in a set containing patches having roughnesses of 20 microinch, 32 microinch, 50 microinch, 80 microinch, and 125 microinch arithmetic-average values (Fig. 7). Although users may desire to check the linearity of their instrument on all the roughness patches, the 20 microinch and 125 microinch surfaces are the most useful for instrument calibration and for measuring the radius of the stylus tip.
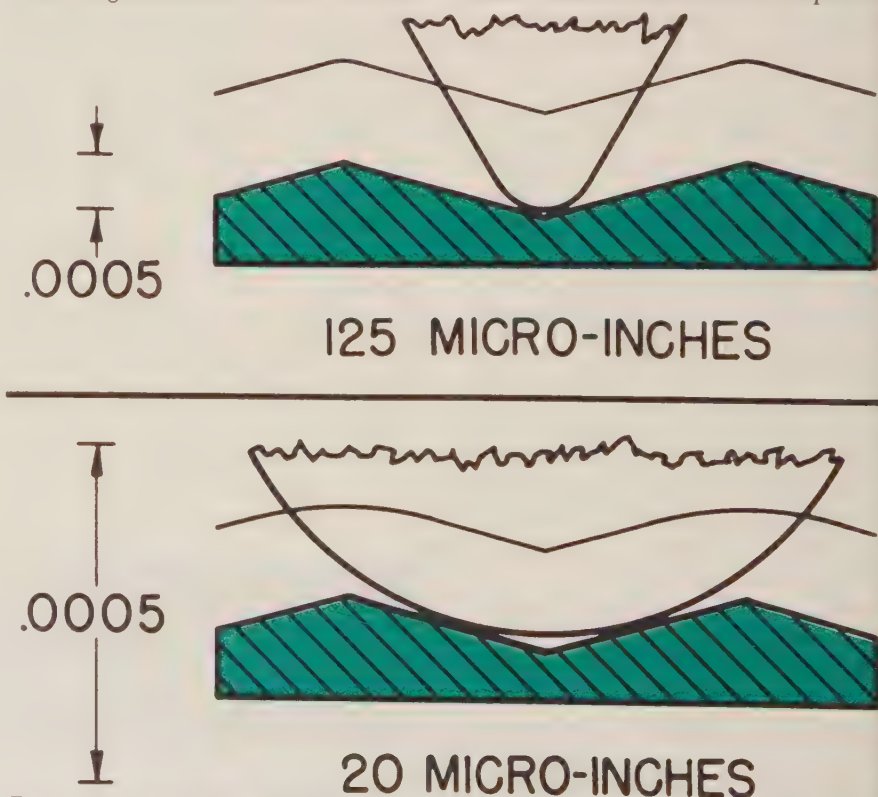
Fig. 8 shows the effect of using a finite stylus radius on the roughness specimen. The reading is 0.3 per cent low for a 0.005 in. radius stylus traversing the 125 microinch surface and the difference is negligible. The lower portion of the drawing shows that the stylus does not penetrate to the bottom of the 20 microinch surface, resulting in a difference of 12 per cent. This scheme is used in measuring the effective stylus radius.

The graph shown in Fig. 9 is typical of the one supplied with each specimen set to aid in taking stylus-tip measurements. The graph is plotted to show the actual tip radius when the instrument reading on the 20 microinch patch is known—assuming, of course, that the instrument is in calibration on the 125 microinch specimen. This specimen set is furnished with each Surfindicator and accounts for its superior accuracy in measuring surface roughness. These reference specimens may be used with any tracer-type instrument to check its calibration and to measure the effective stylus radius.

Conclusion

There are many methods of surface-roughness measurement in wide use throughout industry today, some more accurate than others. As a result of the development of new instrumentation, the control of surface finish has advanced to a point where the designer can specify a certain quality of finish and be confident that the finish can be produced and evaluated throughout the industry.

An Application of Hydraulic Fundamentals: Development of the Hydra-Matic Automatic Transmission

Over the years the gear transmission has served as the conventional device to vary the speed and torque of the rear axle in relation to the speed and torque of the engine. In recent years, the demands of modern traffic have warranted self-shifting transmissions to relieve the driver of gear shifting. Using fundamental hydraulic principles, such transmissions have reduced the actions required in getting a car under way from fifteen movements to three. An automobile in motion tends to stay in motion and the flexibility which hydraulics gives to the designer has enabled the transmission to adjust the engine rpm to rear-axle rpm ratio more favorably for each driving condition than ever before. At Detroit Transmission Division, where the Hydra-Matic automatic transmission is engineered and produced, engineers have coined a new phrase which has been—and remains—the goal of all progress in this advanced application of hydraulic principles. The phrase: *more automaticity*.

THE first automobile transmissions were developed solely to provide performance with little or no consideration given to economy factors. Fuel consumption was of comparatively minor importance in early, low-powered engines but a means of substantially multiplying torque was necessary just to get under way and to climb even a slight grade. Drivers accepted the necessity of shifting gears frequently. With the development of more efficient engines and improved fuels, engineers gave consideration to extracting the reserve-power stores which were dormant in conventional transmissions. By blending certain engineering fundamentals which served as the basic elements of conventional transmissions, engineers have found a way to develop controls which regulate the changeable power delivered by the engine. In addition to the economy benefits achieved, increased safety and driving ease have resulted for the customer. This mechanical device is, in effect, a computer which takes full advantage of the inherent economy of today's engines under widely varying conditions.

Engineering Fundamentals Guided Transmission Engineers

From the very beginning of the automobile, engineers were at work on ways to improve its overall performance. A great deal was accomplished by the steady improvement of the gasoline engine and of its fuel and remarkable gains are still being made in these fields. The means of exploiting these gains were varied but they may be classified roughly into two categories:

- Using a relatively small engine operating at comparatively high speed
- Using a larger engine turning at a lower speed.

The size of the rear wheels (rear-tire rolling radius) and the gear ratio in the rear axle establish the basic N/V relationship in direct drive, N being engine revolutions per minute and V representing the car speed in miles per hour. It is the function of the transmission to change the engine-rpm-to-car-speed relation so the overall speed reduction from engine to wheels permits the most desirable combination of performance and economy under all driving conditions. For a given engine in direct drive, an increase in N/V ratio (obtained by using a high rear-axle ratio) gives better high-gear performance, but only at the expense of fuel economy. A reduction of N/V improves economy but lowers high-gear performance. (There are, of course, high and low limits to N/V beyond which these statements do not hold but they are valid within ordinary design ranges.)

Early Transmission Developments

With the continued improvement in automobiles—and with excessive gear shifting becoming more and more undesirable—engineers and inventors worked on self-shifting devices of many kinds. In spite of this, the only practical relief from frequent gear shifting that the driver obtained for several decades was that due to better high-gear performance. This came primarily from the better—and in some cases—bigger engines and from improved fuels mentioned above. The more powerful engines provided a much greater reserve of power which was available for increased loads. These loads had previously required downshifting of the transmission to attain rapid acceleration and to negotiate moderate grades.

In 1932 a group of General Motors engineers took the first step toward what became the present Hydra-Matic transmission. Their efforts were aimed fundamentally at breaking down the arbitrary limitations on power-train design imposed by the fixed rear-axle ratio. It was clear that, for most ordinary driving, the engine turned over faster than necessary but that drivers could not be expected to go back to frequent manual downshifting, which would be necessary if the rear-axle ratio were reduced appreciably and no other changes made. The engineers set out, therefore, to design a power-train in which the engine would operate at a relatively low N/V under suitable conditions—such as open-road cruising—but which would also provide performance comparable to or even better than conventional transmissions of the day without the use of manual shifting.

Both overdrive and underdrive mechanisms were developed. The underdrive mechanism proved more attractive because it permitted the use of a low rear-axle ratio. A low rear-axle ratio is desirable because it results in low propeller-shaft speed which—as well as being more efficient—results in smoother, quieter operation. One of the underdrive units that appeared particularly promising consisted of a planetary-gear unit mounted behind the regular synchromesh transmission. It was normally locked up in direct drive by a clutch which was applied hydraulically. When the throttle pedal was fully depressed,

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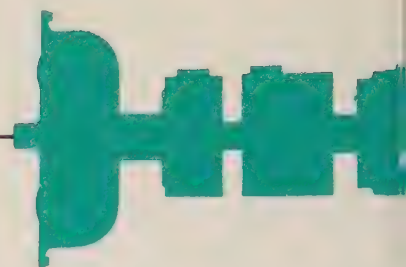
The transmission and rear
axle combination tells the
engine when to work

however, the unit downshifted to under-
drive without torque interruption, thus
increasing the ratio between engine and
rear wheels to obtain higher performance.

This underdrive, which was the fore-
runner of the front planetary-gear unit
of the present Hydra-Matic transmission,
proved that excellent performance could
be maintained without excessively high
rear-axle ratios and without increasing
the amount of manual shifting required.
The drawbacks to the original under-
drive were apparent; it was an extra unit,
added between the conventional trans-
mission (which was retained unchanged)
and the rear axle. The underdrive did
not increase the overall ratio coverage of
the transmission which remained essen-
tially a three-speed transmission with
alternate second gears into which down-
shifts could be made either manually or
automatically.

The next step toward the Hydra-
Matic transmission was a long and im-
portant one. By adding a second plane-
tary unit, the synchromesh transmission
could be eliminated entirely as far as for-
ward speeds were concerned. Moreover,
the transmission had four forward speeds
instead of three: first speed with both
front and rear units in reduction; second
speed with the rear unit only (with rela-
tively high ratio) in reduction; third
speed with only the low ratio front unit
in reduction; fourth speed with both
units locked in direct drive. The front
unit, descendant of the earlier under-
drive, retained its valuable full-throttle
downshift feature to provide surges of
power as needed. All shifts were made
without interrupting engine torque.
Speed-sensitive controls automatically
shifted the front planetary unit in either
of two ranges—low or high. The friction
clutch ahead of the transmission was
retained for starting, but once under
way, a considerable degree of automa-

ticity was obtained.

A semi-automatic transmission which
incorporated the above features was put
into production by General Motors and
used successfully in two car models in
1937 and 1938. However, further devel-
opment on automatic transmissions con-
tinued, with complete automaticity as the
goal. The traditional friction clutch was
replaced by a fluid flywheel clutch to
achieve fully automatic operation and an
entirely new, completely automatic trans-
mission resulted. Under the trade-mark
Hydra-Matic, it was put into production
in 1939 and introduced to the public as
optional equipment on the 1940-model
Oldsmobile. Fig. 1 shows a schematic
diagram which illustrates the principle
of the original and present model Hydra-
Matic-type transmission. While there
have been many mechanical refinements
and improvements in control, the funda-
mental units and power flow remain
essentially the same.

Operating Features

To eliminate shifting the selector lever
to neutral each time the car stops, the
fluid coupling (or fluid clutch) must have
100 per cent slip when the car is standing
with the engine running at idling speed.
To attain this condition the driven torus
must, of course, remain stationary. The
torque which is transmitted by the cou-
pling and multiplied by the reduction
ratio in the part of the transmission fol-
lowing the coupling must be less, there-
fore, than the torque required at the out-
put shaft to start the car from rest. The

torque capacity of a fluid coupling falls
off rapidly as its rotational speed is
reduced, since the torque-speed relation-
ship follows a square curve (Fig. 2). Plac-
ing the coupling, therefore, *after* the front
unit, which has a ratio of 1.45 to 1,
reduces its capacity by the factor
$(1/1.45)^2 = 0.476$, or to less than one-
half the capacity at engine idle speed.
Moreover torque multiplication follow-
ing the coupling is reduced, since the
front unit is not included as it would be
if the coupling were directly behind the
engine. A further reduction in torque is
obtained at this point because the torque
is multiplied only by the rear-unit ratio,
which is 2.63 to 1 and not by both front
and rear-unit ratios as in a direct-coupled
coupling. The total effect, therefore, is
to cut the output torque with the Hydra-
Matic location of the fluid coupling to
approximately one-third at the engine
idle speed. This can be shown as follows:

T_1 = engine torque
T_2 = torque at the coupling
T_3 = propeller-shaft torque.

Therefore, in a direct-drive coupling:
$T_1 = T_2$
$T_3 = T_2 \times 1.45 \times 2.63 = 3.82\ T_1$.

While in the reduction-gear coupling:
$T_2 = T_1 (1/1.45)^2 = 0.476\ T_1$
$T_3 = T_2 \times 2.63 = 0.476\ T_1 \times 2.63 = 1.252\ T_1$.

Thus, it follows that the output-torque
ratio between the two systems is
$1.252/3.82 = 0.328$, or the output torque
of the reduction-geared coupling is 32.8
per cent of the direct-drive coupling.
This permits the use of a high-capacity,
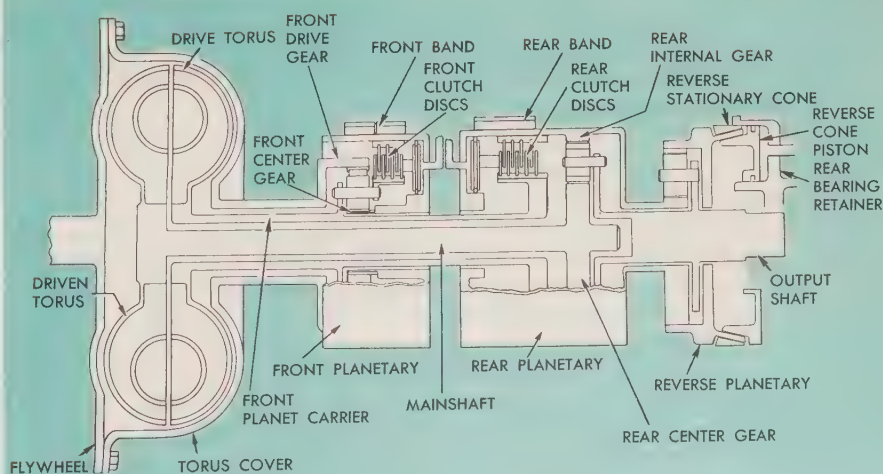efficient coupling without objectionable
creep.

Fig. 1—Schematic diagram of basic Hydra-Matic-type transmission. The principles illustrated
in this model have remained essentially the same as those used on earlier models.
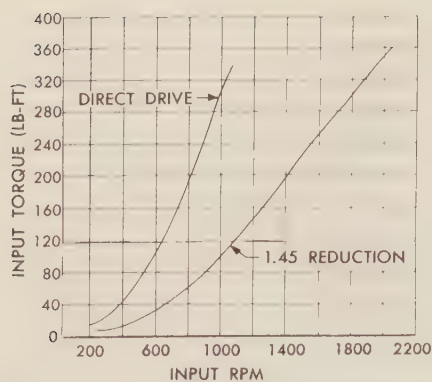
Fig. 2—Square curve (followed by torque-speed relationships) illustrating how rapidly the torque capacity of a fluid coupling falls off after its rotational speed is reduced.

Shift Patterns of the Mechanical Circuit

An important physical difference between the fully automatic Hydra-Matic-type transmission and the earlier semi-automatic transmission was the substitution of the fluid clutch for the pedal-operated friction clutch. Although the fluid coupling is located directly behind the flywheel, the engine torque by-passes it through the torus cover, and the fluid clutch actually follows the front unit in the mechanical circuit. The importance of this can best be explained by following the power flow through the transmission.

In first speed both front and rear units are in reduction; therefore, both bands are applied to hold stationary the front-unit *sun gear* and rear-unit *internal gear*, respectively (Fig. 3). Power flows from the flywheel through the torus cover to the front-unit internal gear, then through the pinions to the front-unit carrier which rotates in the direction of the internal gear at reduced speed (about 7/10 engine speed with a 1.45 front-unit ratio). From the carrier, the power flow moves forward through the intermediate shaft, into the drive torus, through the driven torus to the mainshaft, to the rear-unit sun gear, to the rear-unit pinions and carrier, and finally out through the output shaft.

While in second speed, the power flow is the same as in first speed except in the front unit (Fig. 4). There, the front band is released while the front clutch engages to lock together the planet carrier and sun gear to place the unit in direct drive.

When shifting from second to third speed, a double transition occurs; the front unit again goes into reduction but the rear unit goes into direct drive. The front band is applied, front clutch released, rear band released, and the rear
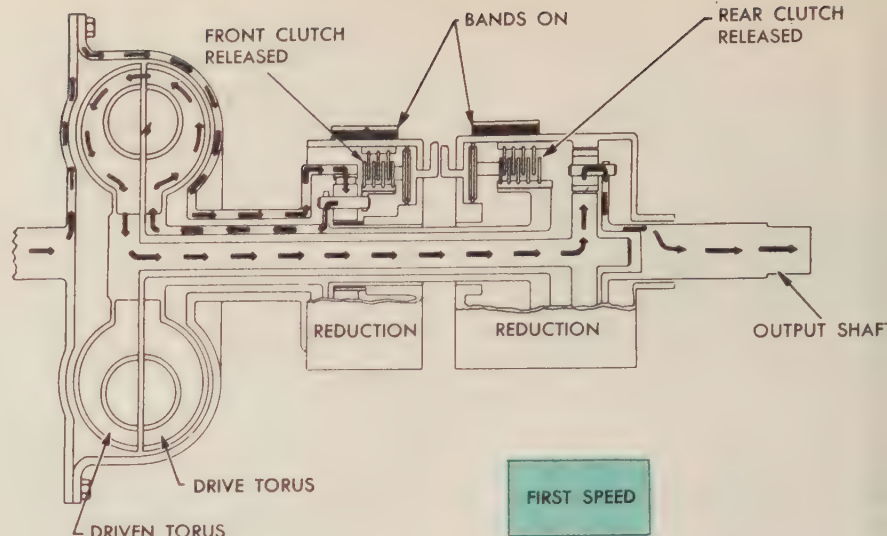
Fig. 3—Power flow in the Hydra-Matic-type transmission under conditions of *first-speed* operation. Both front and rear units are in reduction and both bands are applied to hold stationary the front-unit sun gear and the rear-unit internal gear, not shown in this illustration.
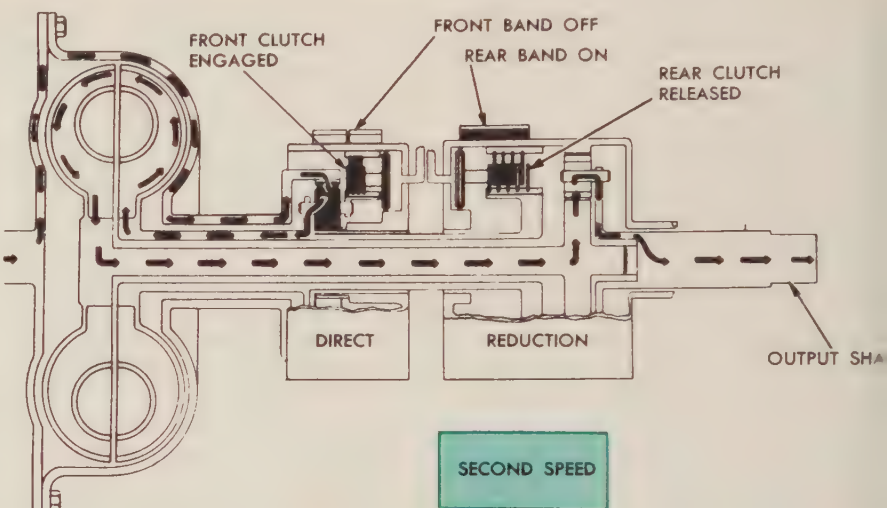
Fig. 4—Direction of power flow under conditions of *second speed* is the same as first speed except in the front unit where the front band is released and the front clutch engaged. This locks together the planet carrier and the sun gear (not shown) to place the unit in direct drive.

clutch engaged. A further advantage of the unique location of the fluid coupling, that is, following the front unit, becomes effective in third speed. The rear clutch does not lock together two of the planet members in the customary manner, but instead, connects the rear-unit internal gear to the intermediate shaft into which part of the output of the front unit flows. Fig. 5 is a schematic drawing of the transmission in third-speed operation and illustrates how power from this shaft follows two separate paths. One moves forward through the fluid coupling and returns through the mainshaft to the rear-unit sun gear. The other path flows directly into the rear-unit internal gear. The pinions in the rear-unit carrier float

between the ring gear and sun gear whose difference in rotational speed is only the small amount of slippage in the fluid coupling. Thus, only part of the torque in third speed is transmitted hydraulically through the fluid coupling—the remainder follows the conventional mechanical path. With this split-torque arrangement, the losses in the fluid coupling, which are already small due to its efficient design, represent an even smaller proportion of the total power transmitted.

In fourth speed the front unit is again locked in direct drive (Fig. 6). The balance of the power flow, including the split-torque feature in the rear unit, follows the same pattern as in third speed. The drive on fourth speed is, of course,
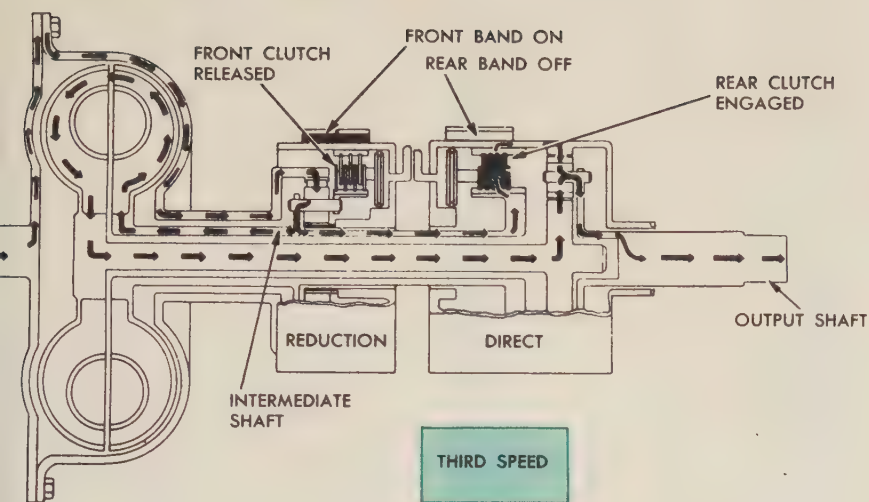
Fig. 5—Diagram illustrating the two separate paths followed by the power flow in *third speed* and the split-torque pattern that develops. One path of power flows forward through the fluid coupling then returns through the mainshaft to the rear-unit sun gear (not shown). The other path of power flows directly into the rear-unit internal gear (not shown). Thus, only part of the torque in third speed is transmitted hydraulically through the fluid coupling, resulting in fewer losses through the fluid coupling of the total power transmitted.
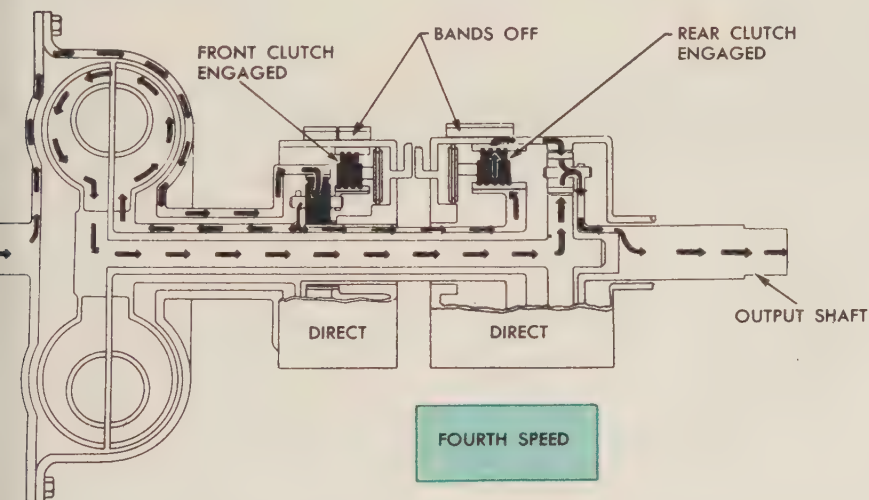
Fig. 6—The above drawing illustrates conditions in *fourth speed* with the front unit again locked into direct drive. Otherwise, the power flow follows the same split-torque pattern as in third speed. Note that drive is straight through the transmission showing no reduction in either front or rear units.

Fig. 7—The above curve illustrates the substantial fuel-economy gains possible by reducing engine speeds and permitting the engine to operate nearer optimum fuel-economy ranges. The vehicle with a 47.8 N/V ratio, obtained with a 3.9 to 1 rear-axle ratio, has an engine speed of 2,868 rpm at 60 mph while, at the same vehicle speed, the car with a 37.7 N/V ratio, obtained with a 3.08 to 1 rear-axle ratio, turns at 2,260 rpm. Such a reduction in engine speed permits the engine to operate closer to optimum fuel-economy ranges.

straight through the transmission without reduction in either front or rear units.

Functional Advantages of the Hydra-Matic-type Transmission

From the above description, it may be seen that the mechanical arrangement of the Hydra-Matic-type transmission offers many important advantages. In all four speeds, it provides ample ratio coverage for excellent performance under any driving condition. It takes advantage of the inherently high efficiency of mechanical gearing in all ratios. It utilizes the good features of a fluid coupling in starting and in its cushioning effect but minimizes the effect of hydraulic losses in the fluid members. Finally, it provides a sure, tight coupling between engine and rear wheels which is equally effective whether the engine is driving the car or the car is overrunning the engine.

The Hydra-Matic-type is, by its nature, an efficient automatic transmission. As pointed out at the beginning, however, it is the overall power-train—engine to rear wheels—that determines fuel economy and performance. Hydraulic controls mate the transmission to its power plant and for this reason, the Hydra-Matic development program has been most affected by the changing characteristics of automotive engines. Mechanical components of the transmission have been redesigned as necessary to increase their capacity or improve their functioning, but the basic mechanical principles have remained about the same.

Operational Controls

A brief description of the operation of the controls will show how the original Hydra-Matic transmission extended the opportunity for improving engine loading. With all shifts automatic, it was possible to rely more on ratio changes and less on excess engine capacity to provide adequate performance in the lower gears, as well as in high gear.

The natural starting point for spacing shift points is the car speed. As the car accelerates at the rate desired by the driver, it should shift upward progressively, and if speed cannot be maintained at a desired level, the transmission should downshift. This immediately introduces a new element in addition to car speed: the speed signal to the control system must be modified to reflect the driver's desire in the matter of performance. Fortunately, this mind-reading job can be readily accomplished mechanically, for the driver naturally indicates his desire to increase, maintain, or reduce speed by the relative position at which he maintains the throttle pedal at a given moment. In the Hydra-Matic transmission, the speed component of the shift originates in a governor driven by the output shaft: it takes the form of oil under a
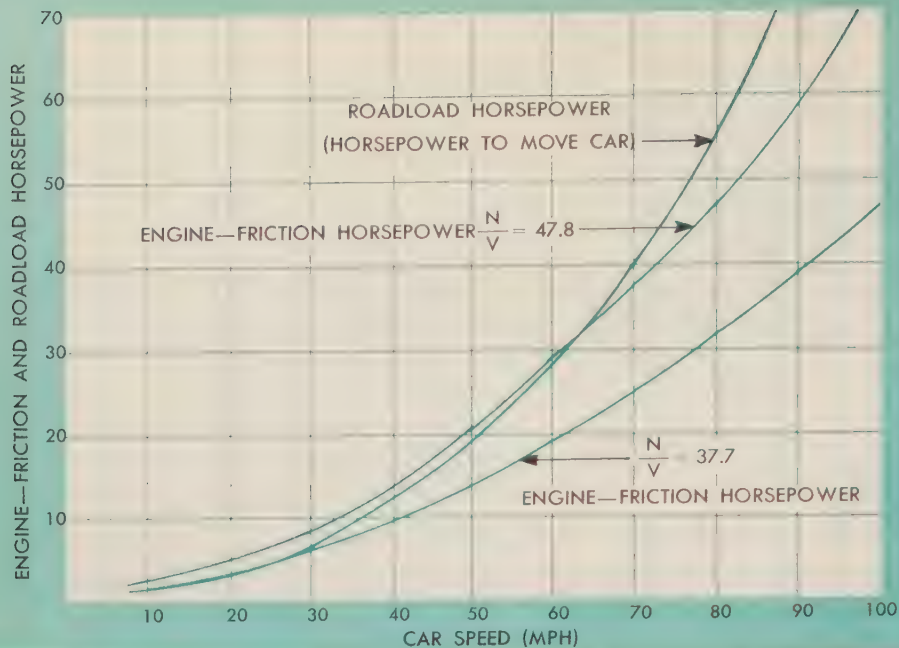
Fig. 8—Shown above is a curve obtained by plotting engine-friction horsepower and road-load horsepower against car speed. Note especially how, with an N/V ratio of 47.8, speeds up to 67 mph use more horsepower to drive the engine than to drive the car.

pressure which is a function of car speed and which acts on hydraulic control valves in a direction to cause upshifts. Upshift movement of the shift valves is opposed by springs, and at very low throttle openings, each shift occurs at the point on the car-speed curve where the force applied by governor pressure overcomes the spring force. As the throttle is opened wider, however, oil is bled through a valve linked to the throttle, and builds up pressure in proportion to the throttle opening. This *TV* (throttle-valve) pressure is applied to the shift valve in a direction which opposes governor pressure, hence adds effect to the spring and raises the shift point.

The described action above is in line with the principle of loading the engine for economy while maintaining performance. A light throttle setting when starting up indicates that the driver does not desire high performance; consequently, the upshifts occur rapidly, with small increments of increased speed. At full throttle, however, the driver is demanding maximum performance; TV pressure is high, and the shift points are held back so they occur at much higher points on the speed curve, with wide increments between shifts. In fact, an upshift will not occur at full throttle until the engine is able to accelerate to where its speed approaches the revolutions per minute at which it delivers its maximum power.

The downshift pattern is similarly affected by speed and throttle position—the transmission stays in high down to very low speed with light throttle, but downshifts more readily as the throttle is opened. As a general rule, Hydra-Matic transmission tends to operate in the highest gear—that is, with the lowest N/V and most favorable engine loading—which is compatible with the driver's demand for performance.

Economy Considerations

Consider now the effect of high and low N/V ratios on economy. Shown in Fig. 7 are two N/V curves. One curve represents an N/V (47.8) curve obtained with a 3.9 to 1 rear-axle ratio. The second curve shows an N/V (37.7) curve obtained with a 3.08 to 1 axle ratio. It can be seen that, at 3,000 engine rpm, a vehicle with an N/V of 47.8 will attain a car speed of 62.8 mph, while a vehicle with an N/V of 37.7 will reach 79.4 mph at the same engine speed. This can be stated in another way. A vehicle with a 47.8 N/V ratio at 60 mph has an engine speed of 2,868 rpm, while at the same vehicle speed, the car with a 37.7 N/V ratio turns at 2,260 rpm. This reduction in engine speed results in substantial economy gains by allowing the engine to operate closer to the best fuel-economy ranges as follows:

In spite of developments which have brought engines up to their present high

state of perfection, the internal-combustion engine is inherently an inefficient mechanism because of high friction and pumping losses. In Fig. 8 engine-friction horsepower and road-load horsepower are plotted against car speed. It can be seen that with an N/V ratio of 47.8 up to 67 mph, it takes more horsepower to drive the engine than is used to drive the car. In other words, the average engine wastes more gasoline in overcoming its own friction and pumping losses than it uses to propel the car. The curve shows that the engine losses which waste gasoline increase rapidly with engine speed, emphasizing the fact that the slower an engine can be run and still produce the required amount of power, the better the economy obtained. Large gains in road-load economy are possible by reducing the engine-rpm-to-car-speed; that is, the N/V ratio.

Curve X of Fig. 9 is a typical engine specific-fuel curve at 2,000 rpm and shows that the amount of fuel used per brake-horsepower-hour is considerably lower when the engine is running near full capacity than when running lightly loaded. From this, it may be seen that the best fuel economy is obtained when running the engine wide open, or nearly so. A common belief is that such a curve shows that an engine burns fuel to produce power most economically when it is running near wide-open throttle. This statement is entirely true when thinking in terms of *actual* delivered horsepower of the engine. An interesting point frequently overlooked, however, is that the engine does not actually burn fuel any more *efficiently* at full throttle than it does at light throttle. The total horsepower produced by an engine can be divided into:

A horsepower used to drive the engine and

B horsepower delivered at the flywheel.

Considering the total horsepower as A plus B, an engine has about the same specific fuel consumption at any throttle opening. However, the difference is that the delivered or usable horsepower with the engine running at full power is a much greater percentage of the total horsepower than when running at light load.

Referring to Z curve in Fig. 9, at 2 hp delivered, friction horsepower of the engine is 17. At this point, the specific fuel consumption is 0.7 lb per bhp-hr
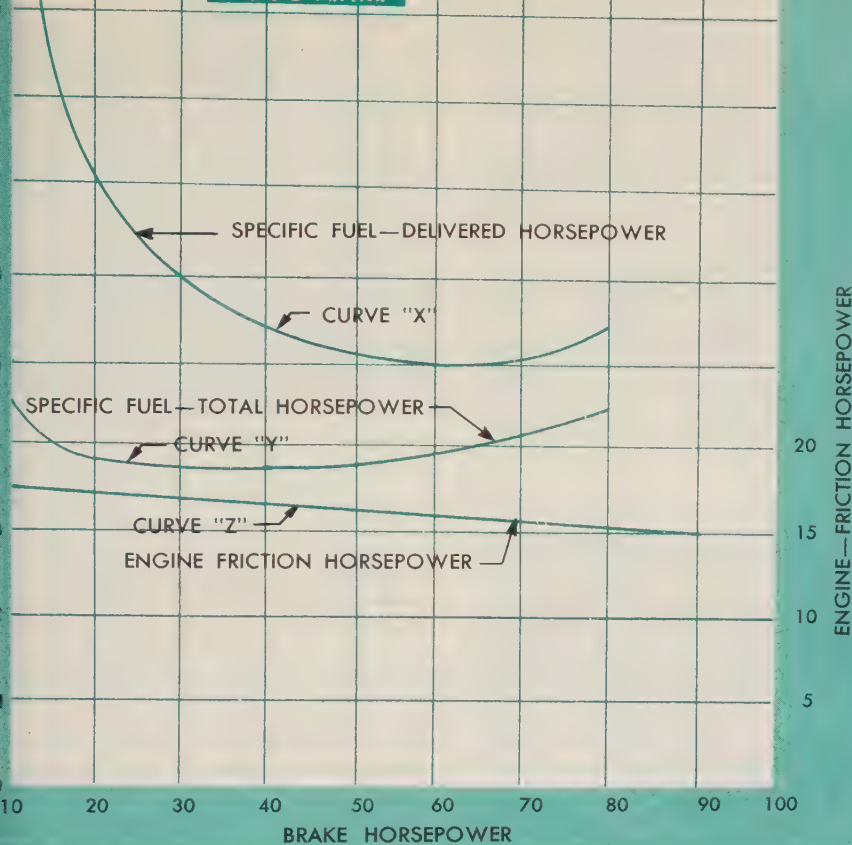
Fig. 9—A typical engine specific-fuel curve at 2,000 engine rpm and at N/V = 37.7. Note that the amount of fuel used per brake-horsepower-hour is lower when the engine is running at full capacity than when it is running lightly loaded.

The total horsepower developed by the engine is 20 plus 17, or 37 hp. Thus, $20/37 \times 0.7 = 0.38$ lb per bhp-hr when considered in terms of horsepower produced to drive the engine plus horsepower delivered.

Taking another point on the same curve, Fig. 9, at 70 hp delivered, the delivered specific fuel is 0.5 lb per bhp-hr. At this point, the friction horsepower is 15.5. Total horsepower of the engine is 85.5 hp or 70 plus 15.5. Thus, $70/85.5 \times 0.5 = 0.409$ lb per bhp-hr, which is approximately the same as the specific fuel at 20-hp output. Curve Y of Fig. 9 is the specific-fuel curve for the total engine horsepower (delivered horsepower plus friction horsepower).

The above data show that gains in fuel economy are possible by operating the engine at as low a speed as possible and under near full power at that speed, because under those ideal conditions the actual friction and pumping losses of the engine are at a minimum compared with the total horsepower produced by the engine.

The advantages to be gained by low engine speeds are not confined to fuel economy, but include smoother and quieter operation, lower oil consumption, and longer life to engine parts.

An important feature of the original Hydra-Matic transmission as it was introduced in 1940 was the use of a lower rear-axle gear ratio. This was made possible by the four forward speeds, wide ratio coverage, efficient operation, and full automatism of controls.

With the introduction of the modern Dual Range Hydra-Matic in 1952 cars, the advantages of lower N/V ratios were still further exploited. While retaining all the features of the original Hydra-Matic transmission, the Dual Range added a second driving position for high performance and an improved shift pattern. As a result of these improvements, cars equipped with Dual Range Hydra-Matic-type transmissions now have the lowest axle ratio ever used in the automobile industry.

Since 1940 when the Hydra-Matic was introduced commercially, over five million have been produced. Twenty-six different models are now being used on passenger cars, from the smallest to the heaviest and most powerful, and on trucks of all types, buses, and military vehicles.

Summary

Almost any transmission can be arranged to work automatically, or partially so; it is primarily a problem of choosing the appropriate controls. This discussion has shown that gains in fuel economy are possible by operating the engine at as low a speed as possible and under near full power at that speed. The early development of underdrive mechanisms with their accompanying low rear-axle ratios lifted the arbitrary limitations on early power-train design imposed by the fixed rear-axle ratio and paved the way to the fully automatic transmission.

With its lower rear-axle ratios and its more efficient use of kinetic energy attained through its unique power-train, the Hydra-Matic-type transmission offers significant economy advantages and justifies the time and money spent in its development.

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Flexible Design of Power Plant Provides Needed Power Now with Minimum Disruption of Overall Plan

By JOHN J. KILMER
AC Spark Plug Division

An expansion program recently undertaken at AC Spark Plug Division's Dort Highway Plant at Flint, Michigan, necessitated the building of a new power plant to take care of the additional steam requirements. Construction on the new manufacturing areas proceeded with such speed that an urgent need for additional steam developed which required that the first part of the new power plant be built to house two of the total of six steam-generating units needed. The overall design of the new power plant is such that the remaining units and their auxiliary equipment can be added with minimum structural changes.

Engineer's task: get 1/3 of ultimate capacity as soon as possible
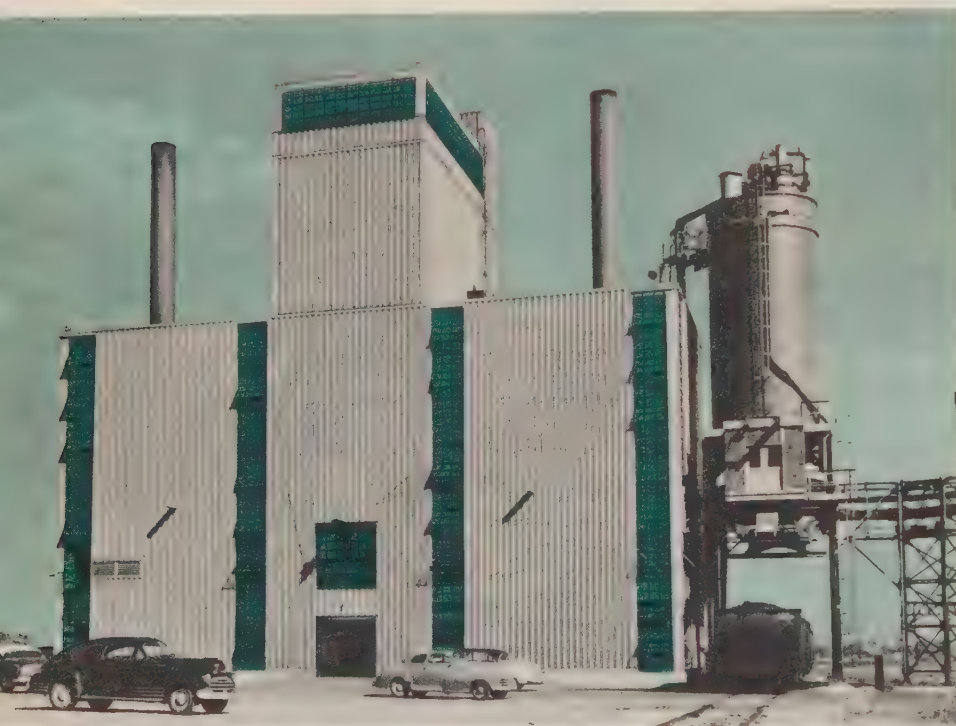
Fig. 1—AC Spark Plug Division's new power plant. The problem was to house two 80,000 lb-per-hr steam-generating units immediately, yet provide easy expansion for four future similar units. Silo at right is of vitrified, glazed tile and is used for storage of ash.

A T AC Spark Plug Division's Dort Highway Plant at Flint, Michigan, steam is used for heating, processing, compressing air, and air conditioning. An expansion program, involving a 50 per cent increase in the plant's manufacturing area, was recently undertaken which required that additional steam be made available to serve the new manufacturing areas. A survey showed that previous increases in the capacity of the existing power plant to a near maximum, together with its confining location with respect to plant buildings, made further expansion in the original location impractical and the need for a new power plant necessary.

The speed with which construction proceeded on the new manufacturing areas caused an urgent need for additional steam and this urgency necessitated early construction of the new power plant. Studies made of immediate loads, anticipated loads due to the expansion program, and expected future loads indicated that a total of six steam-generating units, each of 80,000 lb-per-hr capacity, would be most adaptable to the varying load conditions, both prevailing and expected.

To take care of the immediate steam requirements, the decision was made to have the first part of the new power plant constructed to house two 80,000 lb-per-hr units. This decision was an important factor in the overall design of the power plant in that it was necessary to make provisions for adding the four remaining units and their auxiliary equipment with a minimum of structural changes.

Power-Plant Construction

Acting in an advisory capacity for the new power-plant project was the Power Section of the GM Manufacturing Staff. The drawing up of plans and specifications for the building and equipment was handled by the Argonaut Realty Division, which also supervised construction and installation.

The foundation for the new power plant is unique and is the result of load-bearing soil-test borings, which revealed that unfavorable soil conditions existed. To overcome this situation, it was necessary to drive the 12-in. H-beam foundation pilings to a minimum depth of 20 ft, where load-bearing soil conditions were more favorable. A total of 183 steel piles were used—165 for the building foundation and 18 for the boiler foundations. Groups of the pilings were capped with concrete and regular building procedure started from there. The resulting construction is, in effect, a building within a building.

Construction is of structural steel covered with 3¼-in. insulated, metal siding. Steel sash are arranged in vertical rows on the front and rear. The building is 106 ft wide by 44 ft deep and rises to a height of 114 ft (Fig. 1). A vitrified, glazed tile silo, used for storage of ash, is located

conveniently for ash disposal by either rail or truck. The area surrounding the power plant provides yard storage space for 15,000 tons of coal.

The firing or operating floor of the new power plant is located one floor above ground level. This type of construction provides easy accessibility to ash-collecting hoppers, feedwater-treatment equipment, and maintenance-material storage—all of which are on the ground floor.

Boilers

Each of the 80,000 lb-per-hr boilers now in operation is of the two-drum, vertical, bent-tube-type. The two-hour rating is 90,000 lb per hr. Each boiler has a furnace volume of 3,300 cu ft and an effective heating surface of 9,450 sq ft, which includes a water-wall surface of 1,605 sq ft. The boilers, which do not have superheaters or economizers, are equipped with air heaters having 6,110 sq ft of external heating surface. The allowable working pressure of the boiler drums is 200 psig but the drums are being operated at a maximum of 150 lb of dry saturated steam.

Each boiler now in operation is provided with a 30,600-cfm, forced-draft fan driven by a 40 hp motor and a 52,500-cfm, induced-draft fan with a steam-turbine drive. To facilitate periodic inspection, the fans are at an elevation about halfway between the operating floor and the top of the boilers. Each fan and its drive is mounted on a common steel base which minimizes the possibility of misalignment resulting from conditions of cold starting and warm full-load operation.

The arrangement of air ducts on each boiler for the purpose of conducting experimental work on reinjection and overfire air is a novel feature. By means of sliding gates in the intake ducts, spent gases from the smoke stacks, room air, or air from the air heater may be conducted to the high-pressure fan inlet. This fan's outlet is so connected to reinjection and overfire air inlet nozzles that ash and cinder from collecting hoppers below the last boiler-pass and the air heater, as well as ash and cinder from mechanical dust collectors, is blown into the fire box at the rear of the boiler. Overfire or turbulence air is introduced through nozzles at the front and rear of the boiler (Fig. 2) and at the rear tuyere wall space. Volume passing through each of the systems is
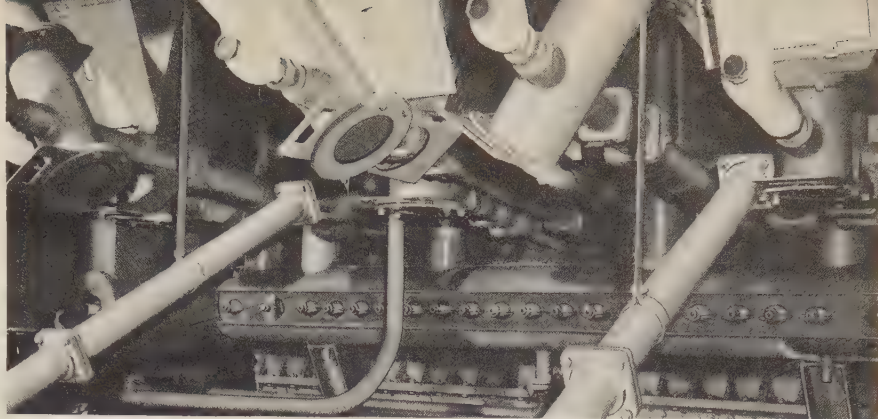
Fig. 2—Arrangement of piping and nozzles at rear of boiler through which overfire or turbulence air is introduced for the purpose of conducting experimental work on reinjection and overfire air.

controlled by dampers in individual ducts.

The boilers are designed with adequate space for installation of heavy-oil and natural-gas burners. The burners have been omitted for the present but full details have been included in the overall design so that installation of the burners can be made without relocating the existing spreader units.

Coal Distribution

The boilers are fired by three-unit spreader stokers with traveling grates that discharge ash continuously at the front. The stokers are designed to burn 12,850 Btu bituminous coal.

The coal bunker, which is located centrally over the firing aisle and above the boilers now in operation, has a capacity of 600 tons or approximately a three-day supply under full-load operation.

From the overhead bunker, coal is discharged from any of four outlets onto a dustless underbunker conveyor of 15 tons-per-hr capacity. The underbunker conveyor's design is such that coal from any section of the bunker may be delivered to each boiler through an automatic, dustproof coal scale which has a rated capacity of 10 tons per hr, with a weigh hopper of 200-lb capacity. Paddle-type switches control the operation of the underbunker conveyor and operate a signaling device if an interruption of any part of the system causes an interruption in the delivery of coal to the boilers.

Additional Features

Approximately 60 percent of the total steam that is generated is condensate that is returned to the boiler. The balance of 40 per cent is made up of raw water obtained from city mains and then

chemically treated. Additional chemical treatment maintains boiler-water alkalinity within desired limits and insures zero oxygen control in the storage tank.

Low pressure steam at 10 psig maintains boiler feedwater close to 240° F. Under normal conditions, a turbine-driven, centrifugal boiler-feedwater pump serves the two boilers but a motor-driven pump is also provided for emergency feed through a secondary piping system. Crossover connections are provided so that either pump may be employed on either feedwater line. Both of the boiler feedwater pumps are located on the operating floor.

To provide for continuous and uninterrupted operation in case of an electrical-power failure, a Diesel-driven, 60-kw, 440-v, 3-phase, 60-cycle a-c generator is installed which provides sufficient power to operate the necessary coal-handling equipment, motor-driven units, and furnishes necessary lighting for individual pieces of equipment as well as general illumination in needed areas.

Summary

As the expansion program of the AC plant nears completion, the four remaining steam-generating units will be installed. Design of the first unit of the power plant is such that the installation of the remaining units can be accomplished with a minimum of structural changes. The urgent need for supplemental steam and the speed with which the first two boilers were brought to completion delayed the installation of metering and control equipment. Even though the two existing boilers need to be operated at partial loads and largely by hand control, the primary task of providing new power when needed has been accomplished.

How Stress Problems
Are Anticipated and Solved
in Automotive Bodies

By WILBUR F. KARBER

Fisher Body Division

Analyzing past stress

data helps to solve

future stress problems

The field of mathematics has many applications in the automobile industry. Of particular importance in the design of a new body model is the advance study of anticipated stress and strength areas of its component parts. This takes place in the design and development stages and involves the compilation of new analytical and test data to be compared with historical data on similar structures. Interpreting stress characteristics by mathematical means thus insures a sound production design. The final verification of such calculations is measured by the customer's criteria of safety, comfort, and performance.
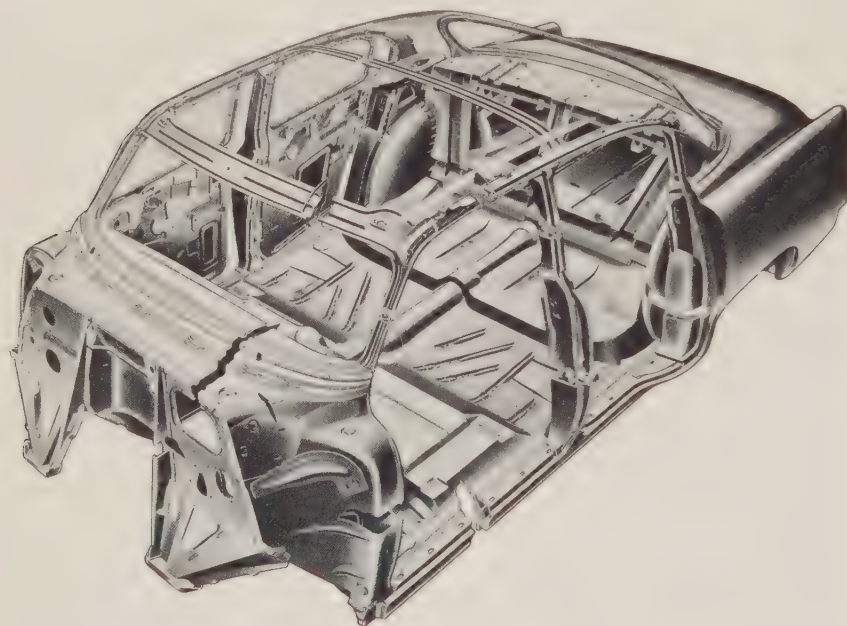
Fig. 1—Diagram of the car body structure illustrating its complex system of panels and fabricated shapes.

THE automobile body, like any other load-bearing structure, must be carefully scrutinized in the development stage to insure that each part is structurally correct. From the beginning of a new body program at Fisher Body Division, as each part of the body takes shape it is reviewed by a body engineer and a stress engineer working jointly to design a body that will be both functional and able to withstand the stresses to which it will be subjected in actual performance.

Stress is generally defined as the resistance of a structural part to distortion. The external forces acting on a structure distort the various component members until the resisting stresses in them are sufficient to maintain the external forces in equilibrium. When an engineer knows the intensity of this resisting stress at any given section of a structure, he is in a position to judge whether this particular part of the structure will stand up under the loads to which it will later be subjected.

The formal design stage of any type of structure, regardless of its usage, follows the same general pattern. First, the external forces and their points of application are determined. Second, the external moments and shears are calculated for all critical points of the structure. Third, the type and size of the various structural elements are decided upon and the stresses calculated. Fourth, the distortion of various critical points of the structure is determined and a decision is made as to whether this distortion is within the limits prescribed for the efficient usage of the structure.

Overall Car Structure

The load-bearing structure of an automobile consists essentially of two separate units—the chassis frame and the body frame. The body is joined to the chassis frame by means of bolts which are installed through oversized holes for ease of assembly. The body is usually separated from the chassis by the use of either tire carcass or rubber shims. These shims are of some importance in helping to deaden the road noise as well as to give smooth-riding qualities to the car. The body and the chassis frame operate as two more-or-less independent structures, each assuming its portion of the car loading according to its respective ability to resist distortion.

The results of comparative tests performed on the body and chassis frame bolted together and on the chassis frame alone have shown that the body is responsible for at least one-half of the car's stiffness.

Body Structure

The exterior contours and dimensions of the body are determined largely on the basis of customer demand and styling considerations. The body engineer designs a body to fit predetermined contours and dimensions. It must be waterproof, dust-proof, and structurally sound. It must operate free of squeaks and objectionable noises under all types of driving conditions. When mounted on the chassis, it must be capable of helping to produce the type of car ride that the engineers have planned for the new
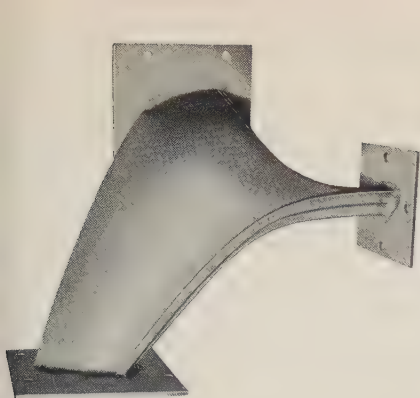
Fig. 2—Fixture used to test the stiffness of stand-ard-size body parts. (Top Left) A windshield upper-corner joint ready to be bolted in the test fixture for a stiffness test. The load is applied by means of a length of pipe and weights sus-pended by a cable. Deflections are measured at the dial locations.

model. A factor of major importance is that the body must be designed so that it can be mass-produced at the minimum cost consistent with the performance required.

The load-bearing structure of the high-production body is a complicated system of comparatively light-gage steel panels and stampings joined by welding proc-esses which have been developed espe-cially for the requirements of high pro-duction. The structure consists of two frames, one on each side of the body. These frames are formed by the dash-to-chassis braces, parts of the front-end sheet metal, front body-hinge pillars, roof rails, center pillars, rockers, quarter-lock pillars, and part of the rear-quarter sheet metal. At the roof level, the frames are joined by a roof panel, windshield header, roof bows, and back-window header. At the floor level, they are joined by a floor pan and a system of cross bars which are bolted either directly to the chassis frame or to outriggers fastened to the chassis frame. A diagram of a body structure is shown in Fig. 1.
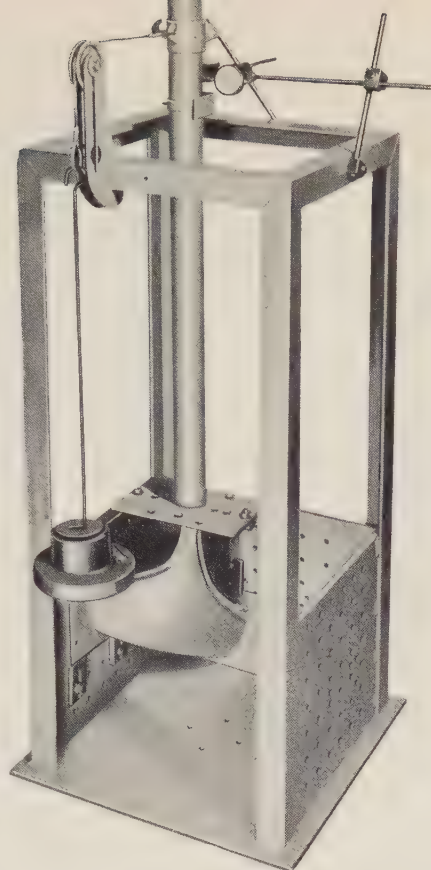
The Effect of Static Load on the Car Structure

The static load supported by the car structure consists of its own weight and that of the passengers and baggage. The car structure must be capable of support-ing this static load while being jacked at either the front or rear bumper without any undue distortion taking place in any of the body openings or panels.

The *critical design loads* for a car struc-ture are those which are created by actual driving conditions. A moving car is con-

tinually subjected to accelerations caused by irregularities of the road surface. The amount and degree of such accelerations depend on the roughness of the road and the speed of the car. Thus, the amount of dynamic loading is usually expressed in multiples of the static load, which is 1g, where g is equal to the gravitational constant. The dynamic loading under driving conditions is de-termined by the use of electronic instru-mentation. Measured dynamic loading over severe bumps at rather high speeds is in the range of 3g.

Designing the Body Frame

At the start of a new body program, the main members of the body frame are designed structurally by comparing the moments of inertia of their typical sec-tions to the moments of inertia of identical sections of past and present production bodies. The size and shape of these new members is normally determined by other than structural requirements. The cal-culated comparison usually establishes the gage of metal and whatever reinforce-ments are required.

Preliminary drawings are first made of the proposed joints of these members;

then a standard test sample is made of each joint and tested in a fixture con-structed for that purpose (Fig. 2). The stiffness of the new joints is then com-pared with that of earlier production joints and the information obtained is used in evaluating the proposed joint design.

The above method is followed through-out the designing of the entire body-frame structure. The structural design of the main frame members is accom-plished by comparing the moments of inertia of typical sections with those of existing bodies, while the joints are checked by comparing their tested stiff-ness with that of identical joints in pro-duction bodies. Table I shows a case example involving the calculation of the moments of inertia of a typical wind-shield-pillar section.

Types of Stress Problems Which Occur in Body Components

Besides the main body frame, many of the body components also present stress problems in their design. A standard for stiffness or strength of the component in question is established by a study of past test data and experience. The component is then designed and finally tested with this standard in mind. Some of the prob-lems encountered are those affecting the hinges, seats, springs, and convertible-top linkages.

Seats

The design of the seat structure is quite important. A well-constructed auto-mobile seat provides the passenger with security and safety while a poorly con-structed one can result in accident and injury. Even if only minor car-seat trouble remained in a seat design re-leased for production, there would be unfavorable customer reaction. The ma-jority of front-seat frames are formed by a system of simple and cantilever beams which are easily analyzed for stresses. The required standards for the strength and stiffness of a front seat have been estab-lished through previous tests, the results of which have been verified by actual usage.

A case example involving the determi-nation of the unit stress in a typical sedan front-seat frame is shown in Table II.

Hinges

Most hinges can be analyzed by the application of fundamentals. Designing
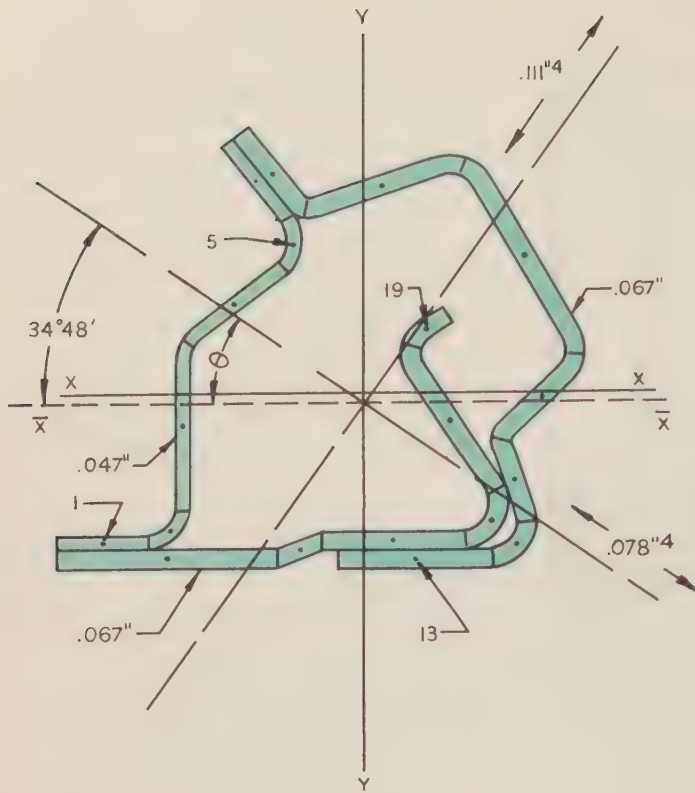
ITEM	AREA			ABOUT X-X AXIS				ABOUT Y-Y AXIS			
NO.	L	T	A	Y	AY	AY²	Ix'	X	AX	AX²	Iy'
1	.30	.047	.0141	-.49	-.0069	.0034	—	-.85	-.0120	.0102	.0002
2	.16	↑	.0075	-.45	-.0034	.0015	—	-.63	-.0047	.0030	—
3	.56		.0263	-.10	-.0026	.0003	.0007	-.58	-.0153	.0089	—
4	.39		.0183	.30	.0055	.0017	.0001	-.41	-.0075	.0031	.0001
5	.17	↑	.0080	.50	.0040	.0020	—	-.23	-.0018	.0004	—
6	.31	.047	.0146	.70	.0102	.0071	.0001	-.35	-.0051	.0018	—
7	.33	.067	.0221	.73	.0161	.0118	.0001	-.29	-.0064	.0019	—
8	.53	↑	.0355	.69	.0245	.0169	—	.06	.0019	.0001	.0008
9	.72		.0482	.45	.0217	.0098	.0016	.51	.0246	.0125	.0005
10	.37		.0248	-.01	-.0002	—	.0002	.57	.0141	.0080	.0001
11	.28		.0188	-.29	-.0055	.0016	.0001	.48	.0090	.0043	—
12	.17		.0114	-.50	-.0057	.0029	—	.48	.0055	.0026	—
13	.50		.0335	-.55	-.0184	.0101	—	.16	.0054	.0009	.0007
14	.72		.0482	-.55	-.0265	.0146	—	-.63	-.0304	.0192	.0021
15	.15		.0101	-.51	-.0052	.0027	—	-.21	-.0021	.0004	—
16	.47		.0315	-.48	-.0151	.0072	—	.10	.0032	.0003	.0006
17	.21	↑	.0141	-.41	-.0058	.0024	—	.40	.0056	.0022	—
18	.56	↑	.0375	-.09	-.0034	.0003	.0007	.27	.0101	.0027	.0003
19	.16	.067	.0107	.21	.0022	.0004	—	.20	.0021	.0004	—
			.4352		-.033	.0143	.0967	.0036	-.009	-.0038	.0829	.0054

$I_{x-x} = .0967 + .0036 - (.4352x - .033^2) = .0999"^4$

$I_{y-y} = .0829 + .0054 - (.4352x - .009^2) = .0883"^4$

$P_{xy} = .0157 - (.4352x - .033x - .009) = .0156$

$TAN. 2\theta = \dfrac{2P_{xy}}{I_y - I_x} = \dfrac{.0312}{-.0116} = -2.690$

$2\theta = -69°-36'$ $\theta = -34°-48'$

$I_p = \dfrac{I_x + I_y}{2} \pm \sqrt{\left(\dfrac{I_x - I_y}{2}\right)^2 + (P_{xy})^2}$

$I_p = .0941 \pm \sqrt{.0058^2 + .0156^2}$

$I_p MAX. = .1107"^4$ $I_p MIN. = .0775"^4$

SYMBOLS

L = LENGTH OF CHOSEN SEGMENTS
T = METAL THICKNESS
A = SEGMENT AREAS
Y = DISTANCE OF EACH AREA FROM THE X-X AXIS
X = DISTANCE OF EACH AREA FROM THE Y-Y AXIS
Ix' AND Iy' = MOMENTS OF INERTIA OF EACH SEGMENT ABOUT ITS OWN CENTROIDAL AXIS
Pxy = PRODUCT OF INERTIA ABOUT THE ASSUMED REFERENCE AXES
Ix AND Iy = MOMENTS OF INERTIA ABOUT CENTROIDAL AXES
Pxy = PRODUCT OF INERTIA ABOUT CENTROIDAL AXES
θ = ANGLE OF THE PRINCIPAL AXES WITH RESPECT TO THE CHOSEN CENTROIDAL AXES
Ip = MAXIMUM AND MINIMUM MOMENTS OF INERTIA

Table I—An example showing how the maximum and minimum moments of inertia are calculated for a given windshield-pillar section (top left). Computations made from the tabular information yield a maximum I_p of 0.1107 in.[4] and a minimum I_p of 0.0775 in.[4].

front- and rear-door hinges for both strength and stiffness is a common stress problem in body design. Hinges must be designed to support the doors properly and yet be no larger than necessary. Excess strength and weight in any body part is expensive because of the large number of pieces usually involved. Hinges must be designed to support the weight of the door plus a design load of 200 lb at the door-handle location. Under this type of loading, the hinges should not contribute more than 0.25 in. to the total door deflection measured at the door handle. Table III is a case example involving the calculation of hinge deflection for an upper hinge in a typical coupe door.

Hinges also are used and stress analyzed in other body locations such as rear-compartment lids, station-wagon lift and tail gates, and sedan delivery back doors. Usually, the most difficult part of the design problem is the determination of the load that the hinge should be designed to withstand and the amount of stiffness that must be designed into the hinge.

Springs

Many types of springs are used in body construction. Cantilever flat springs are used as door checks while coiled springs are used in door-assist and hold-open devices. Tension springs are used in window and seat actuators and compression springs are used as a counterbalance for rear-compartment lids.

The choice of the correct spring for any one of these particular uses is a problem which usually involves consider-
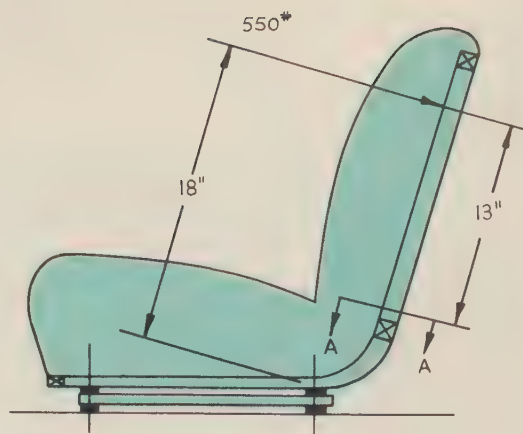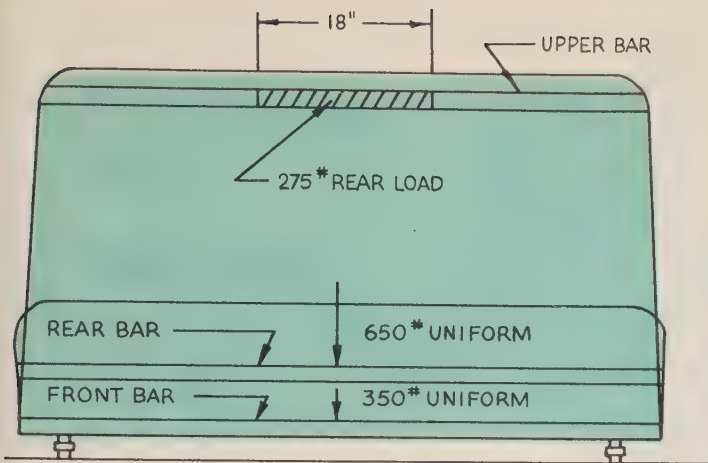
NO.	L	T	A	Y	AY	AY²	Ix'
1	.29	.075	.0218	.52	.0113	.0059	—
2	.28	↓	.0210	.52	.0109	.0057	—
3	1.02		.0765	−.04	−.0031	.0001	.0066
4	.30		.0225	−.65	−.0146	.0095	—
5	.32		.0240	−.72	−.0173	.0125	—
6	.32		.0240	−.72	−.0173	.0125	—
7	.32		.0240	−.67	−.0161	.0108	—
8	1.02		.0765	−.03	−.0023	.0001	.0066
9	.30	↓	.0225	.53	.0119	.0063	—
10	.27	.075	.0203	.53	.0108	.0057	—
			.3331	−.077	−.0258	.0691	.0132

MOMENT OF INERTIA OF SECTION A-A
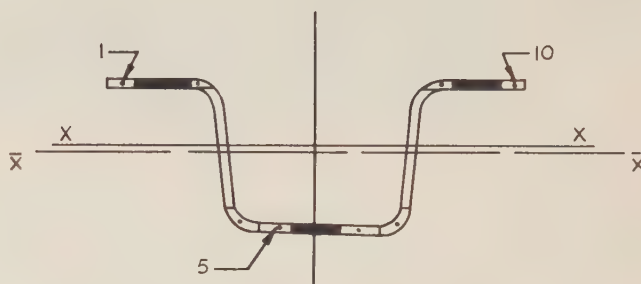
SECTION A-A

$I_{x\text{-}x} = .0691 + .0132 - (.3331 \times \overline{.077^2}) = .0803''^4$

$\dfrac{I}{c}$ (SECTION MODULUS) $= .0803 \div .70 = .1147$

EXTERNAL MOMENT $= 275^\# \times 13'' = 3575''^\#$

$3575''^\# \div .115 = 31,100\ P.S.I.$

DESIGN YIELD STRENGTH $= 32,000\ P.S.I.$

Table II—This case example illustrates the determination of the unit stress in a typical sedan front-seat frame. The various structural members of the seat frame must meet certain design load limits established by past experience. The problem is to determine the unit stress at section A-A in the line diagram of the seat frame (top). The calculated stress in this case is 31,100 psi.

able test work. The difficulty of accurately determining from drawings the amount of work that a particular spring has to undertake permits only a preliminary spring design. For this reason, several test designs are made and tried out in experimental assemblies. The final production spring achieved is the result of this test program. The test program can best be conducted if analytical data are prepared in advance.

A case example illustrating how a stress check is performed on a preliminary design of a door-check spring is given in Table IV.

Folding-Top Linkages

A force analysis is made of most convertible-top linkages and the calculated forces are then used to help design the various parts or links. An estimate of the operational possibilities of a new linkage is obtained by calculating the stalling forces in several positions of the top linkage and comparing them to the calculated stalling forces of existing production tops. From this, it can be determined whether the linkage will operate satisfactorily under a power system equivalent to the one that operated the past production top.

Table V offers a case example showing how the stalling forces working through the convertible-top linkage are calculated for a known piston force when the top is in a stalled position.

Relationship of Welding to the Strength of Body Parts

The structural members of the body are built-up sections comprised of two or more stampings which are joined together by spotwelding. These members are, in turn, connected to each other by spotwelded joints. The number and spacing of spotwelds must be such that the full strength of the members and their joints is developed.

Weld spacing must be such that panels or stamped parts cannot buckle between the welds and joints, and the joint welding must be designed so that the parts framing into the joints are fully connected to each other. The tendency of body openings to deform or *match box* is dependent on the joint construction and welding. Joints and joint welding should be designed to avoid eccentric loading of the welds which would cause the strength of the whole joint to depend on one weld.
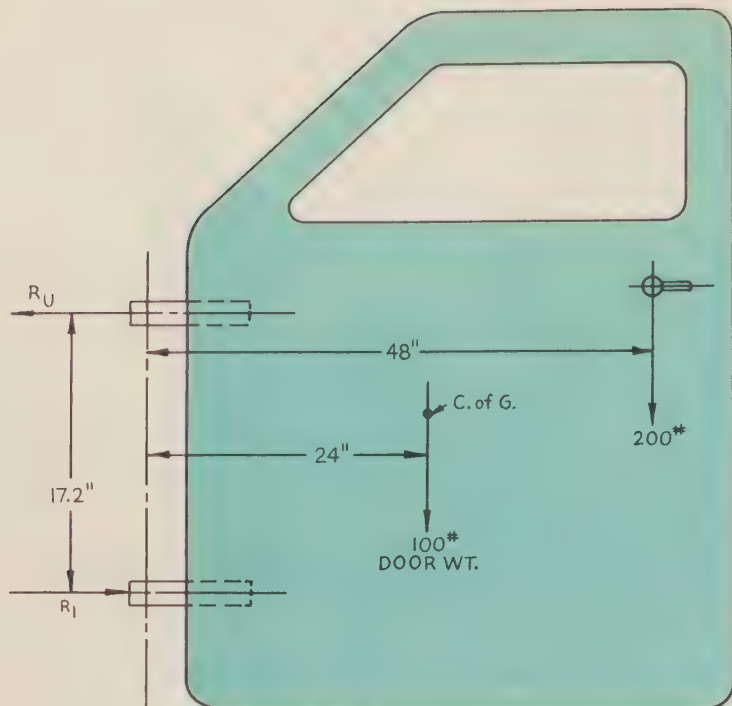
$$R_U = \frac{200^\# \times 48" + 100^\# \times 24'}{17.2"} = 697^\# \text{ HINGE DESIGN LOAD}$$

$$.25" : 48" :: 2\Delta : 17.2$$

$$\Delta = .045" \text{ DESIGN DEFLECTION OF HINGE}$$

SECTION A-A

DEFLECTION $= \dfrac{MDSY}{EI}$

$$= \frac{1,261,302}{29,000,000} = .0435" \quad O.K.$$

UNDERLINE STRESS — SECTION 5½ (F)

$697^\# \times 3.08 = 2150"^\#$

$$F = \frac{2150 \times .25}{.017} = 31,600 \text{ P.S.I.}$$

CORRECTION FACTOR FOR CURVATURE (K)

$K = 1.43$ [ANY STRENGTH OF MATERIALS HANDBOOK]

$F = 31,600 \times 1.43 = 45,200 \text{ P.S.I.}$

HINGE MATERIAL IS S.A.E. 1025 STEEL
DESIGN YIELD $= 41,000$ P.S.I.

THE CALCULATED STRESS OF 45,200 P.S.I. IS HIGH
THEREFORE THE STRAP THICKNESS AT SECTION 5½
WILL BE INCREASED TO .53"

SYMBOLS

DS = ELEMENT OF STRAP LENGTH
T = STRAP THICKNESS
M = PY
E = MODULUS OF ELASTICITY

NO	DS	Y	T	I	M	MDS	MDS÷I	MDSY÷I
1	.70	.35	.50	.0170	244	171	10,059	3,521
2		1.05			732	512	30,118	31,624
3		1.72			1199	839	49,353	84,887
4		2.34			1631	1142	67,176	157,192
5		2.94			2049	1434	84,353	247,998
6		2.94			2049	1434	84,353	247,998
7		2.55			1,777	1244	73,176	186,599
8		2.04			1,422	995	58,529	119,399
9		1.72			1199	839	49,352	84,885
10	.70	1.70			1185	830	48,824	83,001
11	.12	1.70	.50	.0170	1185	142	8,352	14,198
								1,261,302

TABULATION FOR CALCULATING HINGE DEFLECTION

Table III—Calculation of hinge deflection for an upper hinge in a typical coupe door takes into account that the hinge must support 200 lb suspended at the door handle with a maximum allowable 0.25-in. contribution to the total door deflection (top left). The design deflection of either hinge is then calculated as 0.045 in. (Δ). The hinge shown (right) is divided lengthwise into convenient segments (ds) with all dimensions scaled for use in the equations. In this case, the calculated stress was high so the stress engineer recommended increasing the thickness of one part of the hinge, or strap, by 0.03 in.

The use of multiple-electrode press welders for the fabrication of many assemblies complicates the welding problem from the stress angle. The machines are relatively inflexible where spotweld spacing is concerned when compared to the gun-type equipment. The structural necessity of each weld must be carefully reviewed in order not to further complicate the design or interfere with the use of these machine welders.

It is sometimes necessary, because of fabrication or die problems, to design a joint so that some of its mating parts must be joined by rod welding—that is,

S.A.E. 1055
ROCKWELL C-44 TO 48

BRINELL 429-461

1-GROUND
2-MACHINED
3-HOT ROLLED ⎤ FINISH
4-AS FORGED

ALLOWABLE
WORKING STRESS

.55"

.148
DESIGNED DEFLECTION

.28"

1.14"

.047"

.28 R."

P

DS

Y

MAXIMUM WORKING STRESS - 1000 P.S.I.

MEAN WORKING STRESS - 1000 P.S.I.

NO.	DS	Y	L	I	M	MDS	MDS÷I	MDSY÷I
1	.25	.12	.87	.0000075	.12P	.03P	4,000P	480P
2	.25	.37	.94	.0000081	.37P	.093P	11,480P	4,250P
3	.25	.60	1.38	.0000119	.60P	.150P	12,610P	7,570P
4	.25	.86	1.85	.0000160	.86P	.215P	13,440P	11,560P
5	.13	1.03	1.92	.0000166	1.03P	.134P	8,070P	8,310P
6	.13	1.11	1.92	.0000166	1.11P	.144P	8,670P	9,620P
7	.13	1.09	1.92	.0000166	1.09P	.142P	8,550P	9,320P
8	.13	1.00	1.92	.0000166	1.00P	.130P	7,830P	7,830P
9	.40	.74	1.92	.0000166	.74P	.296P	17,830P	13,190P
10	.26	.41	1.92	.0000166	.41P	.107P	6,450P	2,640P
								74,770P

TABULATION FOR CALCULATING DEFLECTION

$$\frac{74,770P}{29,000,000} = .148"$$

$$\underline{P = 57.4^{\#}}$$

MAX. MOMENT = $574 \times 1.11 = 63.7^{"\#}$ @ SECTION 6

$$F = \frac{63.7 \times .0235}{.0000166} = \underline{90,100 \ P.S.I.} = MAX. \ STRESS$$

S.A.E. 1055 - ROCKWELL C 44 TO C 48 (429-461 BRINELL)

DESIGN ULTIMATE STRENGTH = 202,000 P.S.I.
DESIGN YIELD STRENGTH = 183,000 P.S.I.

MEAN WORKING STRESS = 45,000 P.S.I.

MAX. ALLOWABLE WORKING STRESS FOR UNLIMITED LIFE = 72,000 P.S.I. FROM THE ABOVE CURVE. THEREFORE THIS SPRING IS WORKING IN LIMITED LIFE RANGE. THE MINIMUM DESIGN LIFE FOR DOOR CHECK LINKS AND SPRINGS IS 50,000 CYCLES.

Table IV—A case example illustrates how a stress check is performed on a preliminary design of a door-check spring. In this example, the maximum stress is 90,100 psi. If unlimited life for the spring were required, the design would need to be changed to provide a maximum working stress of 72,000 psi or less. The design life for door-check links and springs has been established at 50,000 cycles.

the arc- or gas-type. Due to the gage of the metals involved and the danger of warpage, this type of welding is avoided as much as possible.

Conclusion

The final judge of the structural quality of any car is, of course, the driving public. As a check on analytical studies and generally to ascertain that no structural or mechanical defects exist in a new car model, General Motors submits all new models to a comprehensive test program well in advance of production. As far as the body is concerned, this test program begins at Fisher Body Division as soon as any structural part or sub-assembly—such as a hinge, door, seat, rear-compartment lid—is available for testing. This program then continues until the complete car is given a thorough road test at the General Motors Proving Ground at Milford, Michigan. During this road test, the car is driven over specially constructed roads which subject the body to magnitudes of movement and distortion which are very seldom, if ever, encountered by customer-driven cars.

The resulting body movements and vibrations are recorded by means of electronic instrumentation and all body parts are examined for evidence of failure. The results of these sub-assembly and complete-car testing programs are carefully examined by members of the General Motors Engineering Staff and

150

132

582

2240

600-C

2246-C

2240

582

582

103

1836

2240

132

150#

25"

404

360

463#

2225

2403

478

1996

2.46"

MALE HINGE

1996#

A A

74

D

2440-T

2.84"

810 LB.
℄ PISTON

74#

ALUMINUM ALLOY
CASTING 355-T6
DESIGN YIELD=20,000 p.s.i

FINALIZED MALE HINGE

Table V—The forces working through the convertible-top linkage can be calculated for a known piston force when the top is in a stalled position. In this case example, the forces working at the pivot are calculated when the top is stalled in a full-open position. These forces are used to find the correct size for the male hinge (enlarged, left).

TO FIND: PROPER SIZE OF MALE HINGE AT
SECTION A-A

EXT. MOM. $= 1996^\# \times 2.46'' - 74^\# \times 2.84'' = 4700''^\#$

WIDTH OF SECTION IS SET AT .60 FOR PROPER
CLEARENCES (W)

$F = \dfrac{Mc}{I}$ $\dfrac{I}{C} = \dfrac{4700''}{20,000} = .235$

$\dfrac{WD^2}{6} = .235$ $D^2 = \dfrac{.235 \times 6}{60}$ $D^2 = 2.35$
$D = 1.53''$

serve to develop future design criteria. The stress engineer becomes familiar with the results of these tests and uses them as a basis for future body designing. The entire specialized field of body-stress engineering thus progresses by an evolutionary process in which analytical results are constantly improved through comprehensive and continuing testing programs, both on individual parts and on the entire automobile.

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The New Engineer Has a New Dimension
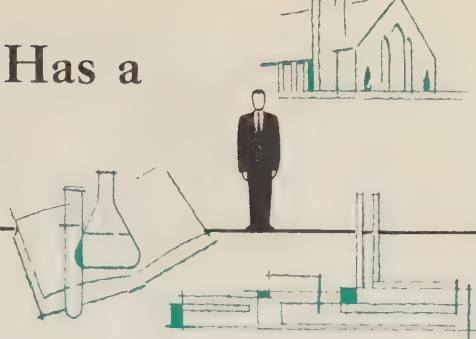
By DR. LOUIS J. CANTONI
General Motors
Institute

A psychologist

looks at

today's engineer

W HAT is an engineer? A facetious definition which has been current among students at various engineering schools in the country is the following:

An engineer is a person who passes as an exacting expert on the basis of being able to turn out infinite strings of incomprehensible formulae calculated with micrometric precision from vague assumptions which are based on debatable figures taken from inconclusive experiments carried out with instruments of problematical accuracy by persons of doubtful reliability and questionable mentality for the avowed purpose of annoying and confounding a hopeless and chimerical group of fanatics referred to, all too frequently, as engineers.

Many reactions are possible to this definition, and most of the reactions, after the first chuckle or two, are not favorable to the engineer. One reaction is that the definition is tragi-comic. Comically, the engineer strikes one as a burlesque fellow whose jocular purpose in life is that of mystifying his colleagues. But tragically, in the pile-up of emasculating phrase upon emasculating phrase, the engineer emerges a man stripped of his singular achievement— his technical competence.

The definition, of course, represents a bit of humor and any student with as intensive a program as an engineer's should resort to humor occasionally, if only to maintain his sense of proportion. Perhaps travesty is healthy when it is a family affair. Yet, the joking of intimates can reflect the actual views of certain critically disposed groups. Some educators, some non-engineer management people, employes, and members of the general public entertain a stereotype of the engineer very much like the one quoted above.

Why does the definition convey an impression unfavorable to the engineer?

It is not because of the humorous approach. The real reason is that the vitalizing influence of the human-relations emphasis has been omitted. Without this emphasis, the engineer's know-how is like a gasoline engine which does not power anything because it cannot be sparked into action.

In the nation's industrialized economy, everyone is the loser when a worthwhile new tool or method fails to be adopted in business or in industry. Yet, many new tools and methods fail to be put into use every day because some expert's approach to his colleagues or other workers did not include an understanding of their individual needs and desires.

The engineer has gained wide recognition for the technical competence which, as a matter of fact, he possesses by training and experience. Beyond this, he is gaining recognition for the human-relations understandings and skills which he has begun to build in. A survey of engineering-school catalogues will show that, since World War II, an increasing number of courses are being offered in humanistic studies and in the social sciences, specifically in psychology. In addition, through enrollment in adult-education programs, whether management-sponsored or not, the working engineer has demonstrated his growing interest in the human-relations aspect of his job.

What, then, is the new engineer? The new engineer E_N is the product of his own technical competence C_T in interaction with varied life experiences L_{VE} which have been energized by psychological insight I_P. Expressed as a formula:

$$E_N = I_P (L_{VE} + C_T).$$

Whether he is at home, at work, or in the social and civic community, insight vitalizes the engineer's day-to-day experiences with people. Insight enables him to discover what is implied in their likes and dislikes, their special abilities and peculiar weaknesses, their desires, their ambitions, their attitudes, and the typical ways in which they attempt to stave off anxiety. As his varied and meaningful life experiences interact with his technical knowledge and skills, the engineer recognizes the importance of the human factor in his efforts to improve design, to employ better methods, to reduce cost, to step up quality, or to locate new markets.

Psychological insight does not operate in a hit-or-miss fashion. There is nothing strange about the fact that, on or off the job, people seek social approval and plan situations and events so that they can feel secure. However, when things do not turn out in accordance with their needs and desires, different individuals have different ways of maintaining their self-esteem. The conscious and systematic application of psychological principles toward understanding people's reactions to stress and frustration very often affords insight into the reasons for their behavior.

But it is not enough for the engineer to try to understand the feelings and the motives of others. A growing understanding of others is dependent upon a growing understanding of one's self. As he applies his knowledge of psychological principles to himself, the engineer develops more and more insight into the why's and wherefore's of his own actions and reactions. This self-understanding, in turn, helps him to perceive with increasing accuracy the effects of his own behavior on the behavior of others.

In summary, the new engineer represents the fusion of his own meaningful personal experiences with his technical competence. Because of this fusion, he can work effectively with colleagues and others toward the solution of some of the world's multi-dimensional engineering problems.

Notes About Inventions and Inventors

By PAUL FITZPATRICK

Patent Section
Central Office Staff

A patent attorney

comments on

his profession

MOST engineers and technical people I have encountered during my career as a patent attorney seem considerably mystified by anything having to do with patents and patent law.

Moreover, my colleagues and I find that we patent attorneys are a mystery to engineers. Some seem to regard us as a peculiar type of lawyer quite unable to give practical advice on such elementary matters as debt collections or tenant's rights.

To others, we may be technical writers of a sort, highly skilled in expanding an engineer's 100-word description into 1,000 words.

We may be thought of, at times, as salesmen who are able to persuade the United States Patent Office to grant a patent on a device in which the originator takes no great pride.

There are very good reasons for regarding us as exceptionally ignorant technical people because the engineers with whom we deal always are far beyond us in understanding of the subjects under discussion.

There really is some truth in each of these theories, for a patent solicitor should be a technical man, a lawyer, a writer, and a salesman. Maybe, if he were *really* good in any of these fields, he shouldn't waste his talents as a patent man. But, if he is fairly competent in each, he might turn out to be a quite capable patent solicitor.

A patent attorney usually starts his training by studying physics, chemistry, or engineering, very likely intending to develop into a creative technical man. Then something happens to turn him aside from the straight and narrow path of professional advancement.

Unlike the scientist, who learns more and more about less and less until he knows everything about practically nothing, our attorney learns less and less about more and more until he knows practically nothing about everything. I submit that the latter course is much the more pleasant.

The second stage in the training process is a course in law. During the three or four years this takes, any heavy burden of technical knowledge previously acquired gradually diminishes. Of course, a law school does not teach much if anything about patents, because a law course is even more general than an engineering course; but it helps in many ways, and most people in patent work are trained in law.

Often, the patent man is learning his profession from experience while attending law school. Many people serve as examiners in the United States Patent Office and take law in the evenings. Others work as assistants in attorneys' offices while attending school.

At any rate, experience is necessary for competence in patent law, just as it is in engineering or in tool making. It may be acquired during or after the schooling, but it is safe to say no one finds that our rather peculiar profession comes naturally to him. It has to be learned, by practice, like playing a horn or sailing a boat.

Having, at last, finished law school (and probably resolving never to attend a class again) our attorney usually takes a bar examination. He can now begin to know less and less about law, for most of the law he has studied in school has nothing to do with patent matters.

Most of what he learns now will be from his clients, the engineers who make the inventions and develop the products which provide work for the attorney.

This, it seems to me, is one of the best aspects of the life of a patent attorney, because it is most interesting to discuss any problem with an engineer who is really interested in it and believes he has contributed something to it. We can sit back and listen to a man who is both an expert and (for the moment, at least) an enthusiast. We may put a question occasionally to be sure we're getting the story straight, and get a fine lecture on some phase of technology.

You may wonder how we stay so uninformed with all this scientific instruction. The answer is that most of us bounce around from one subject to another and cover so much territory that we never know a great deal about the technology of any particular field.

We rely on the technical people to tell us what we need to know about each invention. We can then apply our skill and experience in the handling of patent matters to the solicitation of a patent, the resolution of a question of infringement, or what have you.

Our technical clients should always remember that they must lend their special knowledge and skill to the enterprise to provide, with the patent attorney's complementary experience and skill, the basis for a really good job. Close cooperation is essential.

In General Motors, we are particularly fortunate in the technical resources available to us when needed. I, for one, would like to thank my engineer friends for the real pleasure that my associations with them, and their help, have brought to me.

Being a patent attorney is a lot of fun. One deals with alert and interesting people and often with intriguing and important situations. The work offers variety, challenge, and opportunity for progress.

Patents Granted

On this and the following pages are listed some of the patents granted during the period from February 1 to March 31, 1954. The brief patent descriptions are informative only and are not intended to define the coverage which is determined by the claims of each patent.

• **Francis H. McCormick,** *Frigidaire Division, Dayton, Ohio, for a Domestic Appliance, Nos. 2,668,221, 2,668,222, and 2,668,223, issued February 2.* The Frigidaire Wonder Oven in its final and early forms is the subject of these three patents. The oven includes an upper broil heater and a loop-shaped tubular heater in the bottom of the oven. The chief feature is a

movable third heater having a horizontal oven partition wall beneath it. This may be supported and connected to an electrical receptacle at an intermediate level to provide separate upper and lower ovens, each with its own separate thermostatic control; or, this third heater may be connected immediately above the bottom heater to provide a full-size oven. In this setup, the lower heater and its thermostat are disconnected automatically.

Mr. McCormick is assistant chief engineer at Frigidaire. He joined the Division in 1936 as a range engineer, in 1939 was made manager of the Appliance Engineering Department, and in 1942 was promoted to his present position. In this capacity, he is responsible for the design of Frigidaire products including electric ranges, washers, dryers, water heaters, and other new appliances. This work has resulted in 40 patents. Mr. McCormick received the B.S.E.E. degree from Washington State College in 1915.

● **Brooks H. Short and John W. Dyer,** *Delco-Remy Division, Anderson, Indiana, for an Engine-Starter Control System, No. 2,668,247, issued February 2.* This patent is for a starter control system in which operation of the starting motor is prevented during self-operation of the engine by a relay energized by a separately excited generator.

Mr. Short's biography was published previously on page 56 of the March-April 1954 issue of the GENERAL MOTORS ENGINEERING JOURNAL.

Mr. Dyer's biography was published previously on page 52 of the September-October 1953 issue of the GENERAL MOTORS ENGINEERING JOURNAL.

● **Holden D. Woughter,** *Chevrolet Motor Division, Flint, Michigan, for an Industrial Truck, No. 2,667,985, issued February 2.* This patent is for a fork-lift truck in which the fork lift is retractably mounted so that the overall length of the truck may be decreased to facilitate maneuvering in confined spaces.

Mr. Woughter is warehouse layout engineer in the Planning and Rearrangement Department in the General Motors Parts Division of Chevrolet Motor Division at Flint, Michigan, where he has served since his initial employment as a stock picker in 1936. He was promoted to checker in 1937, to pivot man in 1938, to group leader in 1941, to foreman of the Box Department in 1942, and to his

present position in 1946. During the war Mr. Woughter designed boxes and special crates for the export of military vehicle parts.

● **Edward L. Barcus,** *Guide Lamp Division, Anderson, Indiana, for an Auxiliary Lamp Snap-On Filter, No. 2,668,903, issued February 9.* This patent relates to a transparent, plastic light filter with flanges adapted to snap onto a vehicle trouble lamp to convert it into a danger signal lamp.

Mr. Barcus has been a designer in Guide Lamp's Engineering Department since 1945. He was first employed by the Division in 1935 as a tapping machine operator in small-lamp production. Four patents were granted as a result of Mr. Barcus' work on turn signals and automotive lighting equipment, projects on which he was engaged during the post-war period. Prior to this, he was on special assignment on a confidential ordnance project and was a trouble analyzer for the M-3 sub-machine gun project.

● **Lawrence C. Dermond,** *Rochester Products Division, Rochester, New York, for an Engine-Starting Apparatus, No. 2,668,916, issued February 9.* This invention relates to a starter switch operated by the carburetor throttle, with means to effect closing or opening of the switch only after predetermined movement of the throttle.

Mr. Dermond's biography was published previously on page 56 of the September-October 1953 issue of the GENERAL MOTORS ENGINEERING JOURNAL.

● **Thomas J. Henderson,** *Delco Products Division, Dayton, Ohio, for a Tube-Finishing Machine, No. 2,668,461, issued February 9.* This patent relates to a mandrel for finishing the interior surface of a tube with burnishing surfaces of increasing length from one end to the other of the mandrel to impart a mirror-like finish to the tube.

Mr. Henderson serves as supervisor of equipment and methods in the Equipment Section, Processing Department, Delco Products. He was employed by the Division in 1936 as a process engineer, was promoted to senior process engineer in 1940, to foreman of tools and equipment in 1941, and to his present position

These patent descriptions are informative only and are not intended to define the coverage which is determined by the claims of each one.

in 1947. His current responsibility is the supervision of the selection, purchase, and installation of machine tools and equipment.

● **James W. Jacobs,** *Frigidaire Division, Dayton, Ohio, for an Electrical Switch Apparatus, No. 2,668,884, issued February 9.* This patent discloses a starting and overload relay in which paramagnetic particles are held by magnetic fields to bridge gaps between contacts. The circuits are broken either by weakening of the field or by the current heating the particles to demagnetization.

● **James W. Jacobs,** *Frigidaire Division, Dayton, Ohio, for a Two-Temperature Refrigerating Apparatus, No. 2,672,018, issued March 16.* This patent relates to a refrigerator in which a primary evaporator refrigerates the frozen-food compartment and a dependent secondary system, which has its condenser cooled by the primary evaporator, has an automatically defrosting evaporator refrigerating the unfrozen-food compartment.

● **James W. Jacobs,** *Frigidaire Division, Dayton, Ohio, for a Multicompartment Refrigerating Apparatus, No. 2,672,022, issued March 16.* This is a refrigerator provided with cooling units in which the refrigerated outer surfaces of a freezing container defrost during the idle period of the refrigerating system while a refrigerated plate in the container remains at sub-freezing temperatures.

● **James W. Jacobs and Albert Grooms,** *Frigidaire Division, Dayton, Ohio, for a Testing Apparatus for Thermostatic Switches, No. 2,668,436, issued February 9.* In this device, a spring is yoked to apply a load to the bellows of a thermostatic switch held at constant pressure by ice water. The spring adjustment serves to indicate the temperatures at which the switch will open and close.

Mr. Jacobs' biography was published previously on page 36 of the June-July 1953 issue of the GENERAL MOTORS ENGINEERING JOURNAL.

Mr. Grooms is no longer with the Division.

● **Elbert J. Johnson,** *Packard Electric Division, Warren, Ohio, for a Fuse Holder, No. 2,668,888, issued February 9.* This invention relates to a cartridge fuse container of insulating material, one part of which may be fixed to a support and the other part, which forms a cap, is locked thereon

by a bayonet joint to permit fuse replacement.

Mr. Johnson serves as head of ordnance engineering at Packard Electric. He began his career with General Motors in 1929 when he was employed by Delco-Remy Division, Anderson, Indiana. He advanced through several drafting positions and subsequently was enrolled as a co-op student at General Motors Institute. He was graduated from G.M.I. in 1936. Later, he served as an engineering technical clerk and as a tubing and material specifications engineer at Delco-Remy. He transferred to Packard Electric in 1942 where he did ordnance wiring-harness design. In 1948 he set up the Special Development Section and directed this activity until 1951 when he was appointed to his present post.

● Joseph F. Lash, *Research Laboratories Division, General Motors Technical Center, Detroit, Michigan, for an Anti-Hunting System for Constant-Speed Engines, No. 2,668,921, issued February 9.* This invention relates to a very simple electronic governor or anti-hunting system to maintain an engine driving a load at a substantially constant speed.

Mr. Lash is senior engineer in the Special Problems Department of the Research Laboratories. After earning the B.S.M.E. degree from Michigan State College in 1938 he joined the Research Laboratories in this Department. In 1940 he was granted the M.S. degree in engineering science from University of Cincinnati. This is the first patent granted on the basis of Mr. Lash's work with electronic controls for automatic machine tools.

● Carl F. Petkwitz, *Frigidaire Division, Dayton, Ohio, for a Refrigerator Utility-Drawer Mounting, No. 2,668,423, issued February 9.* This patent discloses a refrigerator cabinet with a storage drawer extending from a side wall intermediate two partitions to provide space above and below the drawer between the partitions for storing low jars.

Mr. Petkwitz serves as an engineer in the Engineering Department at Frigidaire where he is currently engaged in the design and development of household refrigerators and component parts. He received the B.S. degree in mechanical engineering from University of Dayton in 1925. He joined Frigidaire in 1930 as a draftsman. Mr. Petkwitz has had five patents previously granted in the field of refrigerator-cabinet design.
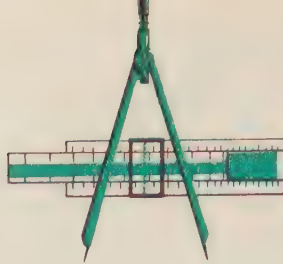
● Raymond E. Schwyn and Ralph H. Mitchel, *AC Spark Plug Division, Flint, Michigan, for a Temperature-Compensator Alloy, No. 2,668,944, issued February 9.* This patent relates to a temperature compensator of negative temperature coefficient of permeability for magnetic measuring instruments.

Mr. Schwyn is a senior research metallurgist at AC Spark Plug. He was first employed as a metallurgist in 1939 after being graduated from Michigan State College with the B.S. and M.S. degrees. At present, he is engaged in the development of the shell-molding process. Previous projects were in the field of temperature-sensitive magnetic materials, which resulted in this patent, and the development of electrodes for cold-cathode discharge tubes.

Mr. Mitchel has been with the AC Spark Plug Division since 1929 when he was originally employed as a laboratory assistant. He is presently engaged in projects involving electron microscopy. Four patents have resulted from his previous work, two in the field of temperature-sensitive magnetic alloys, one covering electronic tubes, and one on engine ignition. Mr. Mitchel earned the B.S.E.E. degree from University of Michigan in 1929.

● Albert P. Dinsmore, Howard M. Geyer, James W. Light, Robert B. Treseder, and Joseph C. Whitmer, *Aeroproducts Operations of Allison Division, Dayton, Ohio, for an Aircraft Propeller-Speed Controller, No. 2,669,312, issued February 16.* This patent relates to an electronic propeller governor for an aircraft. Salient features are the provision of speed sensitivity, acceleration sensitivity, and means for synchronizing a number of engines.

Mr. Dinsmore serves as a project and design engineer in the Engineering Department of Aeroproducts where he is currently engaged in developmental work on supersonic propellers. He originally joined Aeroproducts in 1942 as a vibration tester. In 1943 he was promoted to assistant project engineer and in 1946 assumed his present position. His previous projects have included developmental work on electronic controls. Mr. Dinsmore attended Miami University and was employed by Hamilton Standard Division of the United Aircraft Corporation prior to his joining Aeroproducts.

Mr. Geyer's biography was published previously on page 53 of the September-October 1953 issue of the GENERAL MOTORS ENGINEERING JOURNAL.

Mr. Light is an experimental engineer in the Engineering Department of Aeroproducts. He joined this organization in 1941 as a test operator. In 1943 he was promoted to a special tester and assumed his present position in 1945. He is, at present, engaged in development work on AAHL-A1 actuators (controls) and has previously done work on locking hydraulic actuators. Mr. Light attended Miami University and Ohio State University. His work has resulted in two granted patents, one on a propeller-speed controller and the other on a biasing-type electronic synchronizer.

Mr. Treseder is chief engineer of the Aeroproducts Operations of the Allison Division. He joined Aeroproducts in 1946 as a senior engineer and was promoted to his present position in 1953. He is currently engaged in developmental work on gas-turbine propellers and actuators for aircraft-control surfaces. A graduate electrical engineer from the University of Utah in 1937, his work has resulted in three patents and numerous papers published in connection with his work on propellers, actuators, and electronic synchronizers.

Mr. Whitmer is a senior specifications man in the Engineering Department of Aeroproducts. He originally joined this operation of the Allison Division in 1946 as a special tester and was promoted to his present position in 1950. He is currently engaged in special electronic-controls work and the AG-1 generator after previously doing engineering inspection on electronic controls. Mr. Whitmer attended the Radio and Television Institute of Chicago. His work has resulted in one previously granted patent on a propeller-speed controller.

● Edward P. Harris and Frederick W. Sampson, *Inland Manufacturing Division, Dayton, Ohio, for a Mold and Method of Molding Foam-Rubber Strips and the Like, No. 2,668,987, issued February 16.* This invention relates to an apparatus and

method for continuously forming foam-rubber sealing strips from fluid, foamy, liquid latex in a closed, continuously moving rubber mold that is laterally flexed at two stations to permit injection of the liquid latex and removal of the formed strip after the liquid latex is set or cured in the mold.

Mr. Harris' biography was published previously on page 35 of the June-July 1953 issue of the GENERAL MOTORS ENGINEERING JOURNAL.

Mr. Sampson's biography was published previously on page 35 of the June-July 1953 issue of the GENERAL MOTORS ENGINEERING JOURNAL.

● **Hans O. Schjolin,** *GMC Truck & Coach Division, Pontiac, Michigan, for a Remote-Control System for Gear Shifting in Angularly Disposed Vehicle Transmissions, No. 2,669,316, issued February 16.* This patent discloses a simplified rocking and sliding linkage, which repeats the motion of a driver's gear-shift knob, for shifting the gears of a non-automatic transmission located at the rear of a bus.

Mr. Schjolin's biography was published previously on page 54 of the September-October 1953 issue of the GENERAL MOTORS ENGINEERING JOURNAL.

● **William E. Brill and Arne J. Hovde,** *Cleveland Diesel Engine Division, Cleveland, Ohio, for an Engine Cooling System, No. 2,669,978, issued February 23.* This patent has to do with circulating liquid coolant through the cylinder-wall jacket of a loop-scavenged engine. The coolant is introduced above the exhaust ports and flows first downwardly through the exhaust-port bridges and thence upwardly through the intake-port bridges to the cylinder-head jacket.

Mr. Brill's biography was published previously on page 34 of the June-July 1953 issue of the GENERAL MOTORS ENGINEERING JOURNAL.

Mr. Hovde is a design engineer in the Engineering Department at Cleveland Diesel. He has been engaged in new Diesel-engine design since 1929 with this Division and with its predecessor, the Winton Engine Company of Cleveland, Ohio. He began as a draftsman and advanced through the positions of de-

signer, layout man, senior layout man, and special design engineer. This is the first patent resulting from his work. He was graduated from the Technical University of Norway in 1923.

● **James S. Burge, Hilton J. McKee, and Warren M. Rider,** *Delco-Remy Division, Anderson, Indiana, for an Armature-Coil Lead Staker, No. 2,669,771, issued February 23.* This patent is for an automatic assembling machine for dynamoelectric-machine armatures, particularly for inserting and staking the ends of lead wires into notches in a commutator.

Mr. Burge is no longer with the Division.

Mr. McKee's biography was published previously on page 33 of the June-July 1953 issue of the GENERAL MOTORS ENGINEERING JOURNAL.

Mr. Rider is no longer with the Division.

● **Joseph R. Pichler,** *Frigidaire Division, Dayton, Ohio, for an Open-Top Display Case with Slidable Hood, No. 2,669,851, issued February 23.* This patent relates to a refrigerated open-top display case with a slidable hood on it which permits loading of food into the case over the rear wall while customers may withdraw foods from the case at the front.

Mr. Pichler is section head of the Commercial Engineering Department of Frigidaire. He joined this Division in 1934 as a tracer after receiving the Bachelor of Industrial Engineering degree from Ohio State University. After a series of promotions, which included project engineer on war products, he assumed his present position in 1950. His work in the field of refrigeration has included developmental work on refrigerated display cases and, at the present time, he is concerned with automatic ice-cube makers and with reach-in refrigerators.

● **Virgin C. Reddy and Carlton J. Willrich, Jr.,** *Detroit Diesel Engine Division, Detroit, Michigan, for a Governor Mechanism, No. 2,669,983, issued February 23.* This patent relates to a variable-

speed-type Diesel-engine governor having a manually operable and a speed-responsive fuel-control means interconnected by a floating lever with the engine-fuel regulator, the floating lever being engageable with a smoke stop to prevent overloading and smoking of the engine.

Mr. Reddy serves as a development engineer in the Engineering Department at Detroit Diesel where he is in charge of all basic engine developmental work. Mr. Reddy joined this Division in 1938 as supervisor of the Engineering Laboratory. He received the M.S. degree in mechanical engineering from Iowa State College in 1935 and completed a graduate course in Diesel engineering at General Motors Institute in 1936. His work has resulted in several patents in the field of governors and dual-fuel engines.

Mr. Willrich is no longer with the Division.

● **Paul L. Schneider,** *Delco-Remy Division, Anderson, Indiana, for a Starter Control Apparatus, No. 2,670,444, issued February 23.* This invention is directed to a starter control switch operated by the carburetor throttle shaft. The principal feature is resilient mounting of a means for preventing closing of the switch on a suction-operated diaphragm.

● **Paul L. Schneider,** *Delco-Remy Division, Anderson, Indiana, for an Index-Dial Illuminator, No. 2,671,425, issued March 9.* This patent describes a gear-selection indicator suited for use with automatic transmissions, having a transparent stationary dial bearing indicia and a prism mounted to move behind the dial and transmit light through the indicia.

Mr. Schneider is section engineer of heavy-duty equipment at Delco-Remy, a position he has held since March 1952. Since coming to General Motors in 1921 as a trainee for service engineer, he has worked as process engineer, shop foreman, assistant foreman, supervisor of the Engineering Laboratory, and development engineer. His work on automotive electrical parts has resulted in 16 granted patents in this field. His formal education was obtained at Ohio State University where he received his degree in mechanical engineering in 1921.

● **Wayne H. Sheley,** *Delco-Remy Division, Anderson, Indiana, for an Apparatus for Machining an Internal Cylindrical Surface, No. 2,669,888, issued February 23.* This

These patent descriptions are informative only and are not intended to define the coverage which is determined by the claims of each one.

patent is for a machine for counterboring metal work-pieces in which the operations are automatic and the feed is continuous.

Mr. Sheley is an engineer in the Process Department at Delco-Remy, currently engaged in work on costs and methods analysis. Prior to this assignment, he was occupied with the development of machine design and specifications, a project which resulted in one granted patent. Mr. Sheley's military leave from Delco-Remy was spent in the U. S. Navy. He was separated from the service in 1945 with the rank of lieutenant and returned to Delco-Remy as a senior designer. Purdue University granted Mr. Sheley a degree in electrical engineering in 1930.

• **Raymond Q. Armington,** *Euclid Division, Cleveland, Ohio, for a Swivel Connection for Hydraulic Hoists, No. 2,670,717, issued March 2.* This patent describes a hydraulic cylinder swingably mounted on pipes which supply the cylinder which, in turn, are supported at a distance from the cylinder to make the mounting slightly flexible.

Mr. Armington is general manager of Euclid Division which was recently made a part of the General Motors organization. Mr. Armington began his employment with Euclid as superintendent in 1931 when it was known as Euclid Road Machinery Company. He was made general manager in 1937, president in 1951, and then appointed general manager when the Company became a part of General Motors. Mr. Armington was granted the Bachelor of Industrial Engineering degree from Ohio State University at Columbus in 1928.

• **John O. Almen,** *Research Laboratories Division, Detroit, Michigan, for a Hydraulic Motor-Actuated Marine Propeller-Pitch Control, No. 2,671,518, issued March 9.* This patent is for a hydraulically actuated reversible propeller-pitch control system for heavy-duty applications. The arrangement of the control mechanism and support bearings provides a strong structure for accurately maintaining any selected propeller pitch.

Mr. Almen retired in 1952 after serving for 26 years in the Research Laboratories Division as an engineer and consultant. He joined the Research Laboratories in 1926 as a research engineer in the Power Plant Section and in 1928 was named head of the Dynamics and Gear Sections. In 1938 he became
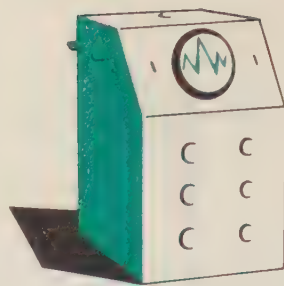
head of the Mechanical Engineering Department No. 1. During his career, Mr. Almen initiated development work on resonance silencing, contributed to a new method of designing gears from the standpoint of fatigue life, did pioneering work in shot peening, and did developmental work on extreme-pressure lubricants. Mr. Almen is an engineering graduate of Washington State College.

• **Max G. Bales,** *Delco-Remy Division, Anderson, Indiana, for an Ignition Timer, No. 2,671,829, issued March 9.* This patent relates to breaker plates for use in ignition distributors. The plate is mounted for rotative movement upon non-metallic bearing elements held by the distributor housing, which reduces friction and breaker-plate wobble.

• **Max G. Bales,** *Delco-Remy Division, Anderson, Indiana, for a Pressure-Actuated Control Mechanism, No. 2,672,890, issued March 23.* This patent deals with an improvement in vacuum motors of the type used for advancing the spark in accordance with engine-suction conditions which increases the life of the motor diaphragm and provides positive stops to limit movement thereof.

Mr. Bales' biography was published previously on page 52 of the September-October 1953 issue of the GENERAL MOTORS ENGINEERING JOURNAL.

• **Engelbert A. Meyer,** *Fisher Body Division, Detroit, Michigan, for an Adjustable Spring Clip for Molding Strips, No. 2,671,254, issued March 9.* This patent pertains to a molding clip which permits both longitudinal and lateral adjustment of the molding strip relative to the supporting surface.

Mr. Meyer is senior project engineer in the Central Engineering Department of Fisher Body. Following his employment with this Division in 1935, he was assigned as a tracer, progressing to detailer, engineer, design engineer, project engineer and, subsequently, to his present position of senior project engineer.

His work on automotive snap-attachments has resulted in four patents.

• **Milton H. Scheiter,** *Detroit Transmission Division, Ypsilanti, Michigan, for a Five-Speed Compound Planetary Gear, No. 2,671,359, issued March 9.* This patent relates to a compact planetary transmission of compound-type having three clutches which can be located in the engine flywheel and can be operated by fluid pressure in a selected pattern to drive various input gears. There are two brakes coordinated with the three clutches to provide three forward drives of reduced speed, direct drive, and reverse.

Mr. Scheiter is section engineer at Detroit Transmission where he is currently engaged in experimental work on future automatic transmissions for passenger and heavy-duty vehicles and power-transfer units for turbo-prop aircraft. Mr. Scheiter received his Bachelor of Mechanical Engineering degree from G.M.I. in 1943 during his employment with Detroit Transmission and, shortly afterward, entered military service, spending three years in the U. S. Navy. This is the first patent granted as a result of Mr. Scheiter's work on automatic transmissions.

• **George R. Bayley,** *Buick Motor Division, Flint, Michigan, for an Automotive Vehicle Hood Edge Bumper, No. 2,672,942, issued March 13.* This patent is directed to arrangements for protecting the external finish on a fender or other portion of an automobile body adjacent a flush-fitting engine hood or other door during its opening and closing movements.

Mr. Bayley is associate staff engineer at Buick Motor. His entire career has been spent at this Division where he began in 1925 as a co-op student from General Motors Institute. Following his graduation from G.M.I. in 1929, he began work as a student engineer and progressed to body draftsman, chief draftsman and, since 1945, has held staff-engineering positions related to body and sheet-metal engineering. Among the many projects undertaken by Mr. Bayley during his nearly thirty years' employment with General Motors has been the establishment of a decimal dimensioning system which was developed while he was chairman of the GM Standards Subcommittee.

• **Richard J. Brittain, Jr. and C. W. Kien,** *Hyatt Bearings Division, Harrison, New Jersey, for a Journal Box, No. 2,672,382, issued*

March 16. This patent discloses an end-thrust construction for a railroad-car journal box wherein a thrust block backed up by a conical rubber washer and a holder is mounted within the end of the journal box.

Mr. Brittain's biography was published previously on page 55 of the September-October 1953 issue of the GENERAL MOTORS ENGINEERING JOURNAL.
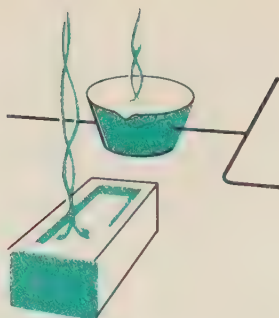
Mr. Kien is design supervisor at Hyatt Bearings, a position he has held at that Division since 1951. His initial employment with General Motors was from 1945 to 1948. He left the organization for a brief period and resumed employment in 1950 as a senior designer. The major portion of Mr. Kien's work has been in the development of journal boxes, a project that has led to three granted patents in this field. Mr. Kien studied engineering at University of Baltimore and at Johns Hopkins University.

• **Eugene B. Etchells,** *Chevrolet Motor Division, Detroit, Michigan, for a Hydraulic Lash Adjuster, No. 2,672,133, issued March 16.* This patent relates to self-adjusting tappets of the telescoping plunger-and-cylinder-type which are side fed with make-up oil from the engine lubricating system, wherein the registering plunger and cylinder inlet ports are arranged to prevent varnish-like deposits between the telescoping surfaces.

Mr. Etchells' biography was published previously on page 53 of the March-April 1954 issue of the GENERAL MOTORS ENGINEERING JOURNAL.

• **Richard S. Gaugler and Edmund F. Schweller,** *Frigidaire Division, Dayton, Ohio, for a Two-Temperature Refrigerating Apparatus, No. 2,672,019, issued March 16.* This patent deals with a refrigerator having a frozen-food compartment insulated from the unfrozen-food compartment in a manner to avoid accumulation of moisture or frost in the insulation surrounding the frozen-food compartment.

• **Edmund F. Schweller,** *Frigidaire Division, Dayton, Ohio, for a Two-Temperature Refrigerating Apparatus, No. 2,672,030, issued March 16.* This patent relates to the construction and arrangement of a refrigerator of the type having a frozen-food

> These patent descriptions are informative only and are not intended to define the coverage which is determined by the claims of each one.

compartment fully insulated from the unfrozen-food storage compartment.

• **Richard S. Gaugler,** *Frigidaire Division, Dayton, Ohio, for a Refrigerator Door, No. 2,673,377, issued March 30.* This patent relates to the door of a household refrigerator held closed by cooperating cylindrical permanent magnets enclosed in cup-shaped iron cladding at the outer upper and lower corners of the door and door jamb.

Mr. Gaugler earned the B.S.Ch.E. degree from Purdue University in 1922 and was elected to the honorary society Phi Lambda Upsilon. He joined the GM Research Corporation, Dayton, in August 1923 where he was made foreman of a pilot plant for blending Ethyl fluid. In 1925 he transferred to Frigidaire Division as foreman-in-charge of process specifications and continued in this capacity until his transfer to the Engineering Department in 1940. Currently, he serves as supervisor of the major product line. A total of 36 patents in the field of refrigeration and air conditioning have resulted from his work.

Mr. Schweller is assistant chief engineer of Frigidaire. He was originally employed by this Division as a draftsman and after serving in this position was promoted to project engineer. Subsequent promotions, prior to his present position, included section engineer and manager of the Household Engineering Department. His technical work at Frigidaire has included developmental work on aluminum-foil insulation and on metal shell and one-piece porcelain refrigerators. He is presently concerned with research and future-product development. This is the thirty-third patent granted as a result of Mr. Schweller's work in the field of refrigeration and air conditioning.

• **John H. Heidorn, James W. Jacobs, and Clifford H. Wurtz,** *Frigidaire Division, Dayton, Ohio, for a Two-Temperature Refrigerating Apparatus, No. 2,672,023, issued March 16.* This invention relates to a

refrigerator in which serially connected evaporators maintain below freezing temperatures in one compartment and above freezing temperatures in another compartment. Means are provided for compensating for any unbalance in load on the evaporators caused by low ambient temperatures.

• **James W. Jacobs and Clifford H. Wurtz,** *Frigidaire Division, Dayton, Ohio, for a Two-Temperature Refrigerating Apparatus, No. 2,672,020, issued March 16.* This invention relates to a refrigerator in which serially connected evaporators maintain below freezing temperatures in a frozen-food compartment and above freezing temperatures in an unfrozen-food compartment. A thermostat on one of the evaporators starts and stops the refrigerant compressor to cause automatic defrosting of that evaporator while maintaining the other evaporator at below freezing temperature.

• **Clifford H. Wurtz,** *Frigidaire Division, Dayton, Ohio, for Plural Refrigerated Compartments with Condensate Disposal Means, No. 2,672,027, issued March 16.* This patent relates to a refrigerator having a frozen-food compartment above an unfrozen-food compartment. A pan and trough receive defrost water from about the frozen-food compartment and direct the water into the unfrozen-food compartment.

• **Clifford H. Wurtz,** *Frigidaire Division, Dayton, Ohio, for a Plural Temperature Refrigerating Apparatus, No. 2,672,028, issued March 16.* This patent relates to a refrigerator construction wherein the freezing- and food-compartment evaporators may be assembled into the cabinet as a unit and then separated and fastened in their final locations.

Mr. Heidorn's biography was published previously on page 45 of the May-June 1954 issue of the GENERAL MOTORS ENGINEERING JOURNAL.

Mr. Jacobs' biography was published previously on page 36 of the June-July 1953 issue of the GENERAL MOTORS ENGINEERING JOURNAL.

Mr. Wurtz is supervisor of the major product line at Frigidaire. After earning the B.S. degree from University of Illinois he joined this Division in 1929. After completing an engineer-in-training program he was made a junior tester and in early 1938 was promoted to senior tester. Later in the same year he advanced to junior engineer and in 1942
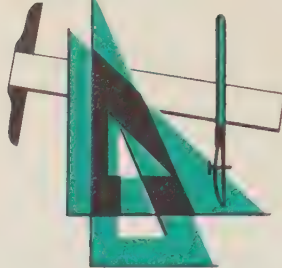
was promoted to senior engineer. In 1950 he was appointed section engineer and in 1952 was promoted to his present position. His work has included the development of a cycla-matic system of automatic defrosting. This is the seventh patent in the field of refrigeration granted as a result of Mr. Wurtz's work.

• **William F. Holin,** *Electro-Motive Division, LaGrange, Illinois, for a Journal-Box End Thrust Construction, No. 2,672,381, issued March 16.* This patent relates to a railroad journal box wherein a thrust block of molded, wear-resistant, antifriction material is backed by a resilient rubber-like cushion and mounted on the end cap of the journal box.

• **Mr. Holin** is a senior project engineer in the Engineering Department at Electro-Motive. He has served with this Division for a period of 18 years, starting as a draftsman in 1936. He advanced through the positions of checker and project engineer before appointment to his present post. The majority of his design projects have been concerned with the development of railroad trucks and the patent described above is the fourth resulting from his work. Mr. Holin obtained his early technical education in Konstanz, Germany.

• **Clarence E. Morphew,** *Cadillac Motor Car Division, Detroit, Michigan, for an Indicating Mechanism, No. 2,672,117, issued March 16.* This patent is on a transmission-selector indicator mounted on the steering column and visible through a window in the hub of the steering wheel. A spot of light and a pointer simultaneously indicate the position of the transmission control.

Mr. Morphew serves as assistant staff engineer in the Body and Experimental Section, Cadillac Motor Car Division. He joined the Division in 1936 as a detail draftsman and has advanced through various engineering positions related to automotive development and to military tank design. His current major project concerns the styling and engineering of new sheet-metal automotive components. Previously, he worked on the initial development of the panoramic windshield. Mr. Morphew attended Lawrence Institute of Technology and Wayne University, both in Detroit.

• **John M. Murphy,** *Frigidaire Division, Dayton, Ohio, for a Two-Temperature Refrigerator having a Special Air-Deflecting Baffle, No. 2,672,026, issued March 16.* This

patent relates to a refrigerator having an improved air-deflecting baffle which serves to prevent condensate from dripping onto the food stored in the lower of two refrigerated compartments.

Mr. Murphy is supervisor of the major product line in the Household Engineering Department of Frigidaire. In 1930 he received the B.S. degree in electrical engineering from Purdue University and immediately joined GM Radio Corporation. In 1933 Mr. Murphy transferred to Frigidaire where he served as an engineer in the Commercial Engineering Department. In 1943 he moved to the Household Engineering Department as a project engineer and served in this capacity until assuming his present position. His work has included design of commercial and household compressor equipment and refrigerating units for household refrigerators.

• **Howard W. Pearsall,** *Research Laboratories Division, Detroit, Michigan, for a Composition for Removing Adherent Deposits from Internal-Combustion Engines, No. 2,672,450, issued March 16.* This patent covers a new liquid composition for removing adherent carbonaceous deposits from metal surfaces such as walls of engine combustion chambers.

Mr. Pearsall is a research chemist at the Research Laboratories Division, a post he has occupied since joining the Division in 1949. Mr. Pearsall has been engaged in chemical research for more than twenty years, including his former associations with the Ethyl Corporation Research Laboratories and Wyandotte Chemicals Corporation. Some of his projects have related to air analyses, exhaust-valve burning, hydrocarbon synthesis and tetraethyl-lead synthesis. He received the B.S.Ch.E. degree from University of Missouri (1931) and the Ph.D. degree from Wayne University (1948).

• **Francis L. Rataiczak,** *Frigidaire Division, Dayton, Ohio, for a Defrosting Refrigerating Apparatus, No. 2,672,021, issued March 16.* This patent is directed to a refrigerator having a frozen-food compartment and an unfrozen-food compartment cooled by two evaporators. An electric heater defrosts one evaporator periodically while the temperature of the other evaporator is maintained substantially constant.

Mr. Rataiczak serves as manager of the Household Engineering Department at Frigidaire. Following his employment in 1926 as a senior test engineer, he advanced to project engineer, section engineer, and ultimately, to his present managerial position. Mr. Rataiczak's work on the development of household refrigerators has resulted in 19 patents granted in the field of refrigeration and ordnance. His formal education was received at Ohio State University where, in 1923, he was granted a bachelor's degree in mechanical engineering.

• **Orson V. Saunders,** *Frigidaire Division, Dayton, Ohio, for a Removable Unit in Refrigerating Apparatus, No. 2,672,029, issued March 16.* This invention relates to a refrigerator having forwardly extending floor-engaging feet increasing the stability of the refrigerator when the door is opened and the food shelves are rolled out.

Mr. Saunders' biography was published previously on page 55 of the November-December 1953 issue of the GENERAL MOTORS ENGINEERING JOURNAL.

• **Sylvester M. Schweller,** *Frigidaire Division, Dayton, Ohio, for a Two-Temperature Refrigerating Apparatus, No. 2,672,025, issued March 16.* This invention relates to the physical relationship between the various elements of a multiple-compartment refrigerator in which two evaporators connected in series maintain the desired temperatures in the freezer compartment and the main food-storage compartment.

Mr. Schweller's biography was published previously on page 55 of the November-December 1953 issue of the GENERAL MOTORS ENGINEERING JOURNAL.

• **William E. Brill,** *Cleveland Diesel Engine Division, Cleveland, Ohio, for a Cylinder-to-Crankcase Bolted Connection, No. 2,672,848, issued March 23.* This patent relates to an engine with mating crankcase and cylinder flanges held together by bolts having

their heads within the crankcase, these heads being retained by L-shaped bolts.

Mr. Brill's biography was published previously on page 34 of the June-July 1953 issue of the GENERAL MOTORS ENGINEERING JOURNAL.

● Elmer J. Hickox, *Packard Electric Division, Warren, Ohio, for a Method of Making Adhesive Tape, No. 2,672,978, issued March 23.* This patent relates to a method for making thermoplastic tape with a surface sufficiently adhesive to sustain the weight of a roll of tape, so that the roll can be readily passed around a wiring harness, by heating the tape to a limited extent while maintained under relatively slight pressure in a roll.

Mr. Hickox is test engineer at Packard Electric. After attending Denison University and Kent State University he joined Packard Electric in 1933 as a laboratory assistant. His technical work has included developmental work on aluminum-wire terminals and high-resistance cables. At the present time, Mr. Hickox is supervising testing work carried out by the Product Engineering Laboratory at Packard Electric. This is the first patent resulting from Mr. Hickox's work in the electrical field.

● George W. Julian and Donald M. Snider, *Delco Radio Division, Kokomo, Indiana, for a Retainer Spring, No. 2,673,334, issued March 23.* This invention relates to a spring supporting structure for radio tubes to maintain them locked in their sockets.

Mr. Julian is senior layout man at Delco Radio where he has been engaged in the design of various radio components. His association with General Motors extends over a period of more than twenty years. In 1933 he began in the blueprint room in the Engineering Department of Delco-Remy Division at Anderson, Indiana. Subsequently, he was advanced through various positions to his present level. This is the first granted patent resulting from Mr. Julian's work.

Mr. Snider is a senior project engineer in the Engineering Department of Delco Radio. He was originally employed at Delco Radio in 1936 as a draftsman and was promoted to project engineer in 1942 and to his present position in 1951. His work has been concerned with the mechanical-engineering phases of automobile-radio design. Mr. Snider earned the B.S.E.E. degree from Purdue University in 1935.

A Typical Problem in Foundry Operation: Determine the Amount of Alloy or Inoculant Required for Specified Properties of Cupola Iron

One of the daily problems of gray-iron cupola operation is the control of the graphite content and the control of other elements added to the base iron to give it the desired properties. Certain of these elements—known as alloys and inoculants—are added to the ladle following the discharge of the iron from the cupola. From the materials available, the foundry engineer frequently must determine the amounts of each to be added to produce the final chemical and physical properties of the iron required by the designer.

By LEONARD W. CZARNECKI

Cadillac Motor

Car Division

How tensile strength

is determined

at the ladle

GRAY iron has a number of characteristics which make it a widely used material for automotive castings. Chief among these are that it is economical, easily melted and cast, easily machined, and it has high compressive strength.

The production of gray iron for automotive applications can be discussed at considerable length. It is sufficient in this instance, however, to recognize certain general principles. Among the factors that affect the structure and properties of iron, the carbon content and the rate of cooling are significant. The problem presented here deals with some of the considerations relating to the carbon content of iron.

Carbon is present in the form of iron carbide (cementite) and free carbon (graphite). The graphite is distributed as flakes or particles throughout a matrix of silicon-bearing iron. The effect of carbon depends upon the amount of total carbon present as well as on the amounts of combined carbon and of graphite. This effect, in turn, is dependent upon the presence of silicon, which element decreases the stability of the iron carbide and promotes graphitization of the iron. Graphitization is inhibited somewhat by the presence of manganese. The distribution and the size of the graphite particles have an important effect on the properties of

the iron; hence, considerable attention is given in the foundry to the control of graphitization through the use of alloying elements in the ladle—a process known as *inoculation*. For a number of reasons, this method of control is more effective than adding the elements in the cupola.

Generally, as the graphite content is increased, the iron becomes softer and weaker but becomes easier to machine, has increased damping capacity, and has better resistance to heat shock. A lower graphite content makes the iron harder and improves the tensile strength, the bending strength, and the torsional strength.

Alloying elements such as nickel, chromium, molybdenum, or copper are often added to cast iron because of the effect of these elements upon the resulting properties of the iron. Chromium, for example, stabilizes the cementite, prevents graphitization, reduces the size of the graphite flakes, and strengthens the matrix. Increased strength and hardness of the iron result.

The problem presented deals with the selection of proper amounts of available foundry materials to be added to the ladle to attain the desired improvements in the iron.

Problem

As it is produced in the foundry, the

Alloy		Per Cent of Alloy Available	Form	Per Cent of Recovery in Ladle
Ferrochrome			20 mesh	85
	Cr	67-70		
	Si	1-2		
V-5 Alloy			20 mesh	85
	Cr	38-42		
	Si	17-19		
	Mn	8-11		
Ferrosilicon			⅜ in. to 12 mesh	95
	Si	83-88		
SMZ Alloy			¼ in. to 32 mesh	95
	Si	60-65		
	Mn	5-7		
	Zr	5-7		

Table I—Typical materials and their compositions which are available for alloying and inoculating with gray iron.

Fig. 1—Cutaway view of the assembled Bui Dynaflow torque converter.

The rotating pump member of the fluid torque converter (Fig. 1) creates a centrifugal pressure which causes a longitudinal separating force which, in turn, tends to separate the pump and the cover. As a result, high tensile stresses are induced in the pump-to-cover bolts. Determination of the tensile stress is dependent upon a basic knowledge of fluid dynamics and calculus. This is the solution to the problem presented in the May-June 1954 issue of the GENERAL MOTORS ENGINEERING JOURNAL. The total longitudinal separating force is 29,290 lb or approximately 15 tons. The stress induced in each bolt is *36,650 psi.*

iron—known as the *base iron*—has certain properties. In this problem the following are considered:

(a) The residual chromium content is 0.10 per cent.

(b) The silicon content is 2.17 per cent.

(c) The base iron has a 163 average Brinell hardness number *Bhn* on a machined surface using a 3,000-kg load.

(d) The base iron has a tensile strength of 30,000 psi.

(e) The capacity of the crane ladle is 1,700 lb.

Typical commercial alloys and inoculants which are available to the foundry are given in Table I. The problem of selecting the proper amounts of materials to be added is a daily occurrence for the foundry engineer and such a problem occurs in various forms.
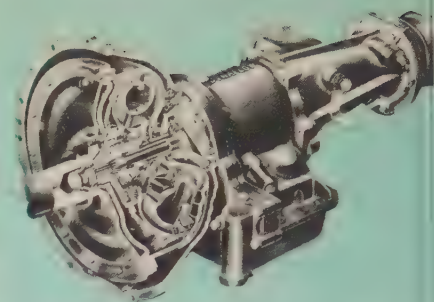
The following are four typical cases:

(a) How much ferrochrome is required to give the iron for cylinder-block castings a chromium content of 0.25 per cent to 0.30 per cent, 197 Bhn to 207 Bhn, and 35,000-psi minimum tensile strength?

(b) For use in a different type casting, such as a cylinder head, how much V-5 Alloy is required to give the iron 0.15 per cent to 0.20 per cent chromium, 179 Bhn to 187 Bhn, and 35,000-psi minimum tensile strength?

(c) In another application to produce iron for miscellaneous castings, how much ferrosilicon is required to give the iron 2.24 per cent to 2.29 per cent silicon?

(d) How much SMZ Alloy is required to give the iron the same silicon content as in (c)?

The solution, which will appear in the next issue, will be a typical one for, as in much foundry work, there are many valid methods of arriving at the correct answer.

THERE are two major forces which produce the longitudinal separating load which causes the pump-cover bolts to be subjected to a high tensile stress. The first force is that due to the centrifugal pressure created by the circulating transmission oil and the second force is that due to the converter-charging pressure. The total of these two forces gives the longitudinal separating load which stresses the pump-cover bolts. The solution to the problem, therefore, revolves about calculating each of the two forces.

Fig. 2 shows the force-pressure distribution in the converter-pump assembly in a high-speed spin. In Fig. 3, an elemental cylindrical ring of fluid is drawn as a free-body diagram showing the equilibrium of forces. This solution to the problem neglects the bending stresses induced in the bolt due to the line of action of the separating force not being coincident with the longitudinal axis of the bolt.

The longitudinal separating force due to the centrifugal pressure of the transmission oil is calculated first. To accomplish this, it is necessary to determine how the centrifugal pressure of the oil

Solution to the Problem: Determine the Tensile Stress in Pump-Cover Bolts on a Torque-Converter Transmission

By ANTHONY J. PANE
Buick
Motor
Division

The bolt tensile
stresses result from
a 15-ton force

increases radially, keeping in mind that at any given converter speed N the centrifugal pressure p is some function of the radius r.

It is first necessary to establish the weight of the elemental cylindrical ring of oil.

Let

ρ = density of transmission oil (0.031 lb per cu in.)
r = inner radius of elemental ring of oil (in.)
dr = thickness of elemental ring of oil (in.)
dh = height of elemental ring of oil (in.)
W = weight of elemental ring of oil (lb).

Then

$$W = \rho \times \text{volume of oil}$$
$$W = \rho\pi[(r + dr)^2 - r^2] \, dh$$
$$W = \rho\pi dh \,[2rdr + (dr)^2].$$

The $(dr)^2$ is negligible. Therefore,

$$W = 2\pi\rho \, dh \, r \, dr.$$

The centrifugal force F_c of the rotating mass of oil is given by the following equation:

$$F_c = \frac{W\omega^2 r}{g} = \frac{(2\pi\rho \, dh \, r \, dr) \, \omega^2 r}{g}$$

where

F_c = centrifugal force of elemental oil ring (lb)
ω = rotational speed of converter (radians per sec)
g = gravitational conversion factor (32.17 ft per sec^2).

Referring to Fig. 3, it is seen that there is an equilibrium of forces existing on the elemental cylindrical oil-ring free body. The forces in equilibrium are the centrifugal force of the transmission oil and an opposing force caused by the product

of the increment of centrifugal pressure and the area of the outer shell of the oil ring on which it acts. Equating these two forces that are in equilibrium will give the following relationship:

$$F_c = dp \times A$$

where

F_c = centrifugal force of elemental oil ring (lb)
dp = increment of centrifugal pressure (psi)
A = outer-shell area of elemental cylindrical oil ring (sq in.).

Substituting into the above equation the value previously calculated for F_c gives:

$$\frac{(2\pi\rho \, dh \, r \, dr) \, \omega^2 r}{g} = dpA.$$

The outer-shell area A of the cylindrical oil ring on which the increment of centrifugal pressure dp acts is:

$$A = 2\pi(r + dr)dh.$$

Substituting the value of A into the above equation and equating the two forces gives:

$$\frac{(2\pi\rho \, dh \, r \, dr) \, \omega^2 r}{g} = dp[2\pi(r + dr)dh]$$
$$dp = \frac{\rho\omega^2 r^2 dr}{g(r + dr)}.$$

Since $r \, (dr)^2$ is negligible, $r^2 dr / r + dr$ will reduce to rdr. Therefore,

$$dp = \left(\frac{\rho\omega^2}{g}\right)rdr.$$

Referring to Fig. 2, it is seen that at radius $r = 0$ the centrifugal pressure is 0 and that at radius $r = r$ the centrifugal pressure is p. Integrating the above expression between these limits gives a general equation for the centrifugal pressure p exerted by the transmission oil.

Integrating

$$\int_{p=o}^{p=p} dp = \frac{\rho\omega^2}{g} \int_{r=o}^{r=r} rdr.$$

Therefore,

$$p = \frac{\rho\omega^2 r^2}{2g}.$$

It is convenient to obtain the above expression in terms of pressure in psi and speed in rpm. This is accomplished by

RADIAL FORCE
RESULTANT
LONGITUDINAL SEPARATING FORCE – F1s
CENTRIFUGAL PRESSURE – P = f(r)
TOTAL PRESSURE – P + Pcc
CONVERTER CHARGING PRESSURE – Pcc

dh
dr
p+dp
P
a
r
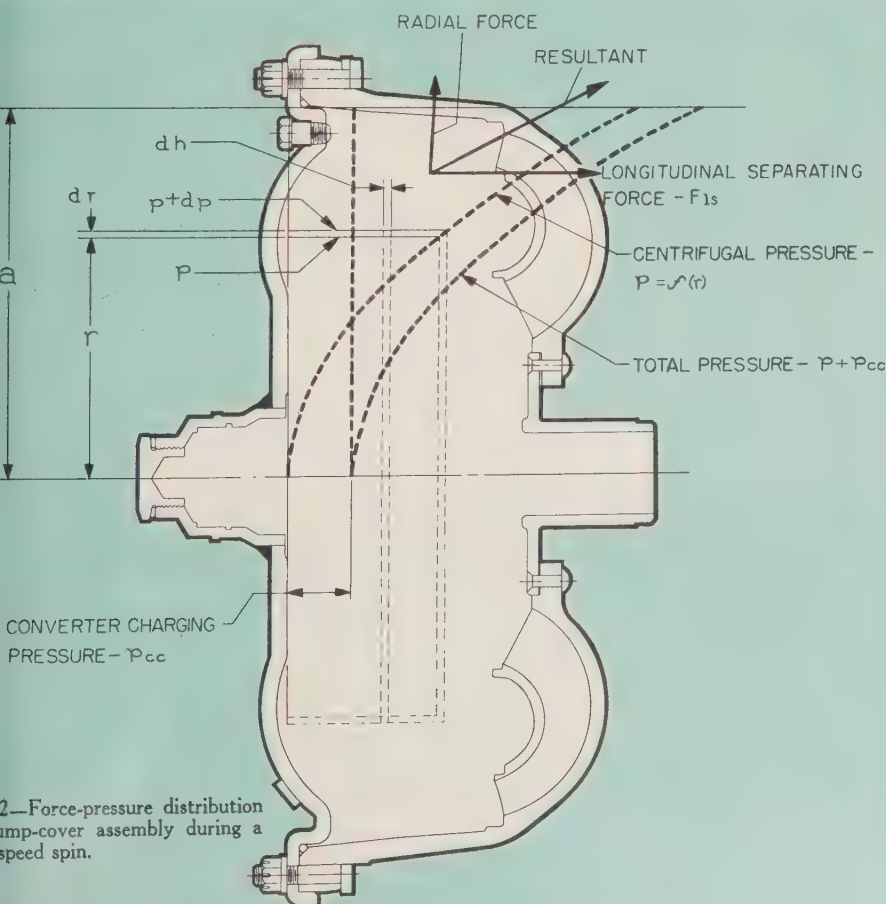
2—Force-pressure distribution [p]ump-cover assembly during a [ ]speed spin.

49
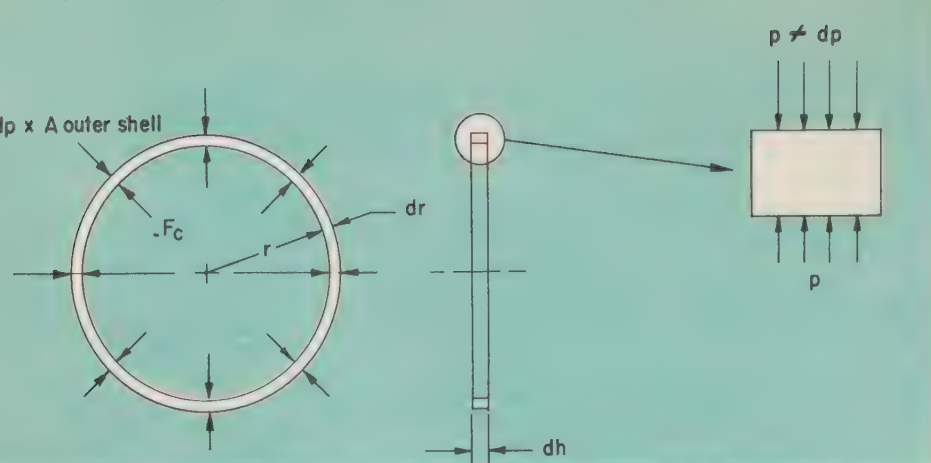
Fig. 3—Free-body force and pressure diagram of an elemental cylindrical ring of fluid of height *dh*.

Fig. 4—The above curve, plotted for values of centrifugal pressure p at various radii r with a constant converter speed of 5,000 rpm, affords a graphical solution to the problem. By multiplying the mean pressure of each segment by each segmental area, the total longitudinal separating force F_{ls} due to centrifugal pressure p of the transmission oil is obtained. Adding this force to the separating force F_{cc} caused by the converter-charging pressure p_{cc} gives the total longitudinal separating force which stresses the pump-cover bolts.
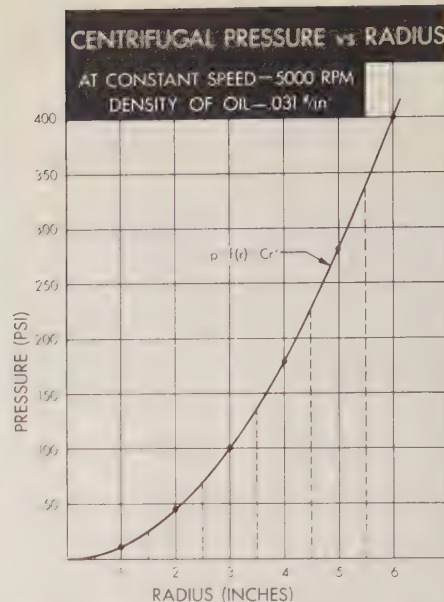
letting ω equal $2\pi N/60$ where N equals the rpm of the converter.

Substituting $2\pi N/60$ for ω in the above equation and the known values for ρ and g will give:

$$p = 0.031\left(\frac{2\pi N}{60}\right)^2 r^2 \bigg/ (2)(32.17)(12)$$

$$p = 0.442 \times 10^{-6} N^2 r^2.$$

The above equation for p is a **specific relationship** for the centrifugal pressure exerted by the transmission oil in terms of converter speed N and radius r for an oil density of 0.031 lb per cu in.

To determine the total longitudinal separating force F_{ls} caused by the centrifugal pressure p of the transmission oil, it is necessary to sum up each elemental cylindrical ring of force. Each elemental cylindrical ring of force will equal the product of the varying centrifugal pressure p and the area on which it acts.

Therefore,

$$F_{ls} = \sum_{r=0}^{r=a} pA$$

where

F_{ls} = longitudinal separating force due to centrifugal force of transmission oil (lb)

A = area on which each varying centrifugal pressure acts (sq in.)

a = maximum radius of pump torus (6 in.).

Solving for A:

$$A = \pi(r + dr)^2 - \pi r^2 = 2\pi r dr + \pi(dr)^2.$$

The $(dr)^2$ term can be neglected.

Therefore

$$A = 2\pi r dr.$$

Substituting into the expression for F_{ls} the above value for A and the previously calculated value of p will give:

$$F_{ls} = \sum_{r=0}^{r=a} 0.442 \times 10^{-6} N^2 r^2 (2\pi r dr).$$

Simplifying:

$$F_{ls} = \sum_{r=0}^{r=a} 0.884 \times 10^{-6} \pi N^2 r^3 dr.$$

The radius varies from $r = 0$ to $r = a$, where a equals the radius of the pump torus. Integrating the above expression between these limits and solving for F_{ls} gives:

$$F_{ls} = 0.884\pi N^2 10^{-6} \int_{r=0}^{r=a} r^3 dr$$

$$F_{ls} = \frac{0.884\pi N^2 10^{-6} a^4}{4}.$$

Substituting into the above equation the known quantities of $N = 5,000$ rpm and $a = 6$ in. gives $F_{ls} = 22,500$ lb, which is the separating force due to centrifugal force of the transmission oil when operating at 5,000 rpm.

In addition to the longitudinal separating force caused by the centrifugal force of the circulating transmission oil, there also is a longitudinal separating force caused by the converter-charging pressure. This force is equal to the product of the converter-charging pressure and the area upon which it acts. The longitudinal separating force F_{cc} due to the converter-charging pressure is:

$$F_{cc} = 60\pi(6)^2 = 6,790 \text{ lb.}$$

The total longitudinal separating force

inducing the tensile stress in the pump-cover bolts is the sum of the separating force due to the centrifugal force created by the transmission oil and the separating force due to the converter-charging pressure or:

Total longitudinal separating force
= 22,500 lb + 6,790 lb = 29,290 lb.

The longitudinal force which acts on each of the 15 bolts is equal to 29,290 lb/15 or 1,953 lb. The tensile stress to which each bolt is subjected is equal, therefore, to the force on each bolt divided by the cross-sectional area of the bolt at the minor diameter:

Mean Centrifugal Pressure p of Segment (psi)	Segmental Area (sq in.)		Longitudinal Separating Force F_{ls} of Segment (lb)
1	π (0.5^2	)	0.78
5	π (1 − 0.5^2	)	11.78
10	π (1.5^2 − 1	)	70.7
34	π (4 − 1.5^2	)	186.8
55	π (2.5^2 − 4	)	388.5
83	π (9 − 2.5^2	)	716.
117	π (3.5^2 − 9	)	1,193.
157	π (16 − 3.5^2	)	1,850.
202	π (4.5^2 − 16	)	2,695.
252	π (25 − 4.5^2	)	3,760.
308	π (5.5^2 − 25	)	5,070.
368	π (36 − 5.5^2	)	6,650.
		Total	22,592.56 lb

Table I—Tabulation of the total longitudinal separating force F_{ls} caused by the centrifugal pressure p of the transmission oil calculated from the graph of Fig. 4.

Bolt tensile stress $= 1,953 / \frac{\pi}{4} (0.2603)^2$

$= 36,650$ psi.

A graphical solution to the problem of determining the bolt tensile stress may also be used. This graph can be used to substantiate the analytical computations or may be used, in itself, as a solution.

Presented in the problem was a table which contained values of centrifugal pressure p at various radii r when the pump was rotating at a constant speed of 5,000 rpm. The given values for p and r, when plotted, result in a graph similar to Fig. 4 which serves the purpose of obtaining a graphical solution.

In Fig. 4 the area under the centrifugal pressure versus radius curve has been divided into segments. By multiplying the mean pressure of each segment by each segmental area, it is possible to do graphically what has been done mathematically when the area under a known curve is integrated between two limits. The product of each segmental mean pressure and each segmental area gives the longitudinal separating force. When the forces obtained from each segment are added together, the longitudinal separating force F_{ls} due to the centrifugal pressure p of the transmission oil is obtained. Adding this separating force to the longitudinal separating force F_{cc} caused by the converter-charging pressure p_{cc} gives the total longitudinal separating force which causes the tensile stress in the bolts.

The accuracy of a graphical solution is dependent upon the accuracy with which the graph is drawn. In Table I is a tabulation of forces calculated from the graph of Fig. 4. The values shown should closely approximate those obtained from plotting the graph on $8\frac{1}{2}$ in. by 11 in. graph paper.

The value for the longitudinal separating force obtained from the graphical solution, when added to the separating force due to the converter-charging pressure, gives a total longitudinal separating force of 29,382 lb. The tensile stress in each bolt, using this value, equals *36,730 psi* which closely compares to the mathematically calculated value of *36,650 psi*.

The solution to the problem points out that the mathematics and analysis used is not overly complicated. Basic integral calculus and dynamics were needed to accomplish the solution, as is often the case with engineering problems.

Recent Speaking Engagements Filled by GM Engineers

Engineer-lecturers, speaking before technical societies and in engineering classrooms, are but one facet of General Motors' continuous program aiming at making available information on GM engineering developments to interested persons outside the organization. Listed below are some of the recent speaking engagements of GM engineers.

Willard R. Houser, supervisor of the Dynamometer Department, AC Spark Plug Division, Flint, discussed "Spark Plugs and Electrical Components" during the Motor Vehicle Maintenance Conference held at the University of Washington, Seattle on March 23.

Robert A. Hard, engineer in the Defense Engineering Department of AC Spark Plug, Flint, completed five speaking appearances during the period covered in this report.

On April 3 Mr. Hard addressed the Michigan Society of Anthropology at Ann Arbor, Michigan, outlining "The Anthropological Aspects of Michigan Bogs."

Mr. Hard appeared before the Mott Foundation Class in Flint, Michigan, on three occasions. On April 6 he spoke on "Astronomy;" on April 13 he discussed "Anthropology;" and on April 20 he spoke on "Mineralogy."

On April 21 Mr. Hard appeared before the U. S. Naval Reserve in Flint, Michigan. The topic of his talk was "Simultaneous Equations."

R. F. Holmes, standards engineer in the Engineering Department of AC Spark Plug, Flint, appeared before the American Society of Tool Engineers at Lansing, Michigan, on April 26. The title of his talk was "Screw Threads."

On May 5, before the Kiwanis Club of Linden, Michigan, **Lloyd F. Christensen,** superintendent of methods and layout at AC Spark Plug, Flint, presented the talk "Problem Solving."

Joseph A. Anderson, general manager of AC Spark Plug, addressed the American Society for Engineering Education at University of Michigan in Ann Arbor. The title of his talk, given on May 8, was "Industry Needs Creative Engineers."

"The Future Electronic Technician in the Engineering Field" was the talk presented to the graduating class of Pulaski High School in Milwaukee, Wisconsin, on April 14. The speaker was **Earl Close, Jr.,** project engineer at AC Spark Plug, Milwaukee, Wisconsin.

O. P. Prachar, head of the Turbo-Jet Engines Research Section of the Engineering Research Department of Allison Division, addressed the National Meeting of the Society of Automotive Engineers held in New York City on April 15. The title of his talk was "Protection of Turbine Inlet Ducts."

"Problems Involved in High-Altitude Flight" was the talk given at University of Kansas, Lawrence, on April 12. The speaker was **Earl Chester,** test pilot at the Kansas City plant of the Buick-Oldsmobile-Pontiac Assembly Division.

Fred H. Cowin, assistant staff engineer at Cadillac Motor Car Division, spoke on "Building Cars as You, a Motorist, Would Build Them" at the March 27 meeting of the Auto Maniacs, an automobile enthusiasts' group, in Detroit.

Harold G. Sieggreen, chief engineer of Central Foundry Division, Saginaw, spoke on "Shell Molding" before the American Foundrymen's Society, meeting in Detroit, Michigan, on April 15.

On May 11 **F. J. McDonald,** factory manager at Central Foundry, discussed "Gating to Control Pouring Rate and its Effect Upon Casting" before the Foundrymen's Convention, meeting in Detroit.

On March 25 **W. T. Burwell,** director of special vehicle engineering at Chevrolet Motor Division, described "How Industrial Suppliers Can Best Help Design

Engineers" at the Industrial Marketing Conference held in Detroit, Michigan.

"The Corvette Plastic Body" was the topic discussed by E. J. Premo, assistant-chief engineer-in-charge of the Passenger Car and Truck Body Design Group at Chevrolet Motor, and presented to the St. Louis Section of the Society of Plastic Engineers and the Downtown Kiwanis Club on March 25.

E. N. Cole, chief engineer and E. H. Kelley, general manufacturing manager of Chevrolet Motor, discussed the "Relationship Between Design Engineering and Production" before the S.A.E. National Production Meeting and Forum held in Chicago, Illinois, on March 30.

Max M. Roensch, director of laboratory tests in the Laboratory Test Group of Chevrolet Motor, addressed the Indianapolis Section of the S.A.E. The title of his talk, given on April 8, was "Automotive Power—Gas Turbines or Piston Engines."

On April 8, before the Sports Car Club of America meeting in Detroit, Z. Arkus-Duntov, assistant staff engineer in the Research and Development Section of Chevrolet Motor, presented a discussion of "The LeMans and its Meaning to the Automotive Industry."

On May 3 Mr. Arkus-Duntov addressed the Southern California Section of the S.A.E. The topic of his talk was "Fundamental Engineering Considerations in the Design of Passenger, Sports, and Racing Cars—A Comparison."

Mauri Rose, experimental engineer in the Vehicle Development Group of Chevrolet Motor, spoke before the Cornell Club of Michigan in Detroit on April 20. He discussed "The Effect of Racing on Automobiles and People."

"The Corvette Plastic Body" was the title of the talk presented to the American Society of Body Engineers, meeting in Detroit on April 23. The speaker was S. C. Richey, assistant staff engineer of Chevrolet Motor's Passenger Car Body Section.

C. W. Frederick, assistant chief engineer, Production Section of Chevrolet Motor's Engineering Department, outlined "What Engineering Management Expects of the Engineer" before the faculty and student body of Virginia Polytechnic Institute on April 23.

Maurice Olley, director of the Research and Development Section of Chevrolet Motor, discussed "Some Merchandising Facts Behind the Corvette" at the Merchandising Clinic sponsored by the American Marketing Association and held in New York City on April 28.

Kenneth A. Meade, director of the Educational Relations Section of the Central Office Public Relations Department, on May 11 addressed the Toledo Engineers Club on the subject "The Engineer and His Profession."

Eric R. Brater, assistant chief engineer in the Engineering Department of Cleveland Diesel Engine Division, appeared before the Society for Experimental Stress Analysis in Cleveland, Ohio, on March 31. His topic was "An Excursion into Space."

On May 12, Ernest A. Jackson, process engineer at Delco Products Division, spoke to the Patterson Vocational High School students in Dayton, Ohio. The title of his talk was "Paint Chemistry—Its Manufacture and Production Methods of Application."

M. G. Wright, head of the Mechanical Engineering Section of the Engineering Department, Delco Radio Division, addressed the Purdue University Student Branch of the Institute of Radio Engineers on March 31. The title of his talk was "The Latest Developments in Automobile-Radio Tuning Techniques."

"The Theory and Practice of Diesel Engine Operation" was the title of the presentation made by Hans M. Gadebusch of the Engineering Department of Detroit Diesel Engine Division. The talk was given on April 27 before the Pittsburgh Section of the S.A.E.

C. H. Palmer, chief metallurgist in the Engineering Department of Diesel Equipment Division, addressed the American Society of Lubrication Engineers in Cincinnati, Ohio, on April 5. The topic of his talk was "The Reclamation of Industrial Petroleum Products."

On April 20 Edward Orent, superintendent of Aeroproducts Activities at Diesel Equipment, and Stuart Kutsche, senior engineer at the same Division, appeared before the senior engineering students at Michigan State College. They discussed "The Design, Development, and Manufacture of a Turbo-Jet Fuel-Burner Nozzle."

L. E. Simon, chief metallurgist in the Engineering Department at Electro-Motive Division, addressed the Pueblo Chapter of the American Society for Metals, Pueblo, Colorado, on March 18. His topic was the "Selection and Specification of Steels."

On March 19 Mr. Simon delivered this same talk before the Rocky Mountain Chapter of the A.S.M. in Denver, Colorado.

H. L. Rittenhouse, manager of the Product Engineering Department at Euclid Division, completed two speaking engagements during the S.A.E. Earthmoving Industry Conference held at Peoria, Illinois.

On April 13, Mr. Rittenhouse outlined "The Need for and Objectives of a Test Code."

The next day he discussed "The Engine Requirements for Earthmoving Equipment."

Louis F. Held, chief product design engineer at Euclid, addressed the senior students at Euclid Senior High School in Cleveland, Ohio, on April 1. In connection with the Annual Vocational Conference Career Week he discussed "The Career Opportunities in Drafting."

Howard K. Gandelot, engineer-in-charge of the Vehicle Safety Section of the General Motors Engineering Staff, GM Technical Center, spoke before the Massachusetts Safety Council Conference held in Boston on March 23. His topic was "Problems and Progress in Automobile Design Safety."

Lyle A. Walsh, manager of the GM Engineering Staff Activities, outlined "Engineering in General Motors—Yesterday and Today" before the Detroit Section of the American Society of Mechanical Engineers, meeting at the General Motors Technical Center on April 13.

On April 14 **John Dolza**, engineer-in-charge of the Power Development Section of the GM Engineering Staff, spoke at Michigan State College, East Lansing on the subject "How to Design an Engine for an Automobile."

Charles A. Chayne, vice president in charge of the GM Engineering Staff, spoke to a Masonic Lodge in Detroit on April 15. The title of his talk was "Automobiles."

On April 29 Mr. Chayne described "Car Engineering in General Motors" to the students at University of Michigan in Ann Arbor.

At the Massachusetts Institute of Technology in Boston, on May 3, Mr. Chayne described "How a New Automobile Comes Into Existence."

"Plastics" was the topic of the talk given on April 21 at the Case Institute of Technology, Cleveland, Ohio, by **C. H. Jensen**, manager-in-charge of the Parts Fabrication Section of the GM Engineering Staff.

Before a Philadelphia, Pennsylvania meeting of the A.S.M.E. on April 21, **George Smith**, assistant engineer-in-charge of the Transmission Development Section of the GM Engineering Staff, discussed the question "Are We Being Taken for a Hydraulic Ride?"

Morris D. Thomas, welding engineering instructor in the Science Department at General Motors Institute, Flint, spoke at the Spring Technical Meeting of the American Welding Society held in Buffalo, New York, on May 5. The title of his talk was "Plant Welding Training Programs."

Erik H. Halvarson, instructor in the Mechanical Engineering Section of the Product Engineering Department, General Motors Institute, presented the paper "Developing Creative Ability through an Engineering Curriculum" before the Mechanical Engineering Conference sponsored by the Michigan Section of the American Society for Engineering Education and held at University of Michigan in Ann Arbor on May 8.

At the Marquette University Annual Engineering Alumni Seminar held in Milwaukee, Wisconsin, on May 1, **John J. Cronin**, vice president in charge of the General Motors Manufacturing Staff, presented the talk "An Adventure: Entering the Higher Pursuits of Life."

Kenneth A. Stonex, head of the Technical Data Department at the General Motors Proving Ground, Milford, Michigan, spoke to the Bureau of Highway Traffic at Yale University on April 21. The title of his talk was "The Relation of Automobile Characteristics to Traffic Engineering."

Before the American Society of Lubrication Engineers meeting in Cincinnati, Ohio, on April 5, **Richard J. Brittain, Jr.**, assistant chief engineer, Railroad Section, Product Engineering Department, Hyatt Bearings Division, described "Roller Bearings for Railroad Equipment."

C. F. Becher, senior engineer in the Product Engineering Department of New Departure Division, discussed "The Application of New Departure Precision Ball Bearings to Machine Tools" at the E. I. duPont Company in Camden, South Carolina, on April 27.

On May 6 Mr. Becher presented the same talk before the American Society of Tool Engineers meeting in St. Louis, Missouri.

D. C. Perkins, experimental engineer in the Product Engineering Department at Oldsmobile Division, appeared before the Hi-Twelve Club of Lansing, Michigan, on March 19. His topic was "What We Are Doing at the Proving Ground, and Why."

Elliott M. Estes, assistant chief engineer in the Product Engineering Department at Oldsmobile, spoke on "Oldsmobile Air Conditioning" before the Evansville, Indiana Section of the American Society of Refrigerating Engineers meeting on May 4.

Robert W. Truxell, methods engineer at Oldsmobile, discussed "Cost Reduction through Pre-Production Planning" before the Industrial Management Society meeting in Chicago on May 13.

Before the Ohio Section of the American Society for Engineering Education meeting at Ohio State University in Columbus on May 1, **George A. Delaney**, chief engineer of Pontiac Motor Division, described "The Engineering of a New Model Car."

"Application of the Two-Beam Interference Microscope to the Study of Surfaces" was the title of the talk given

before the Optical Society of America meeting in New York City on March 25. The speakers were **William L. Grube**, supervisor, and **Stanley R. Rouze**, research physicist—both of the Physics and Instrumentation Department, Research Laboratories Division.

Joseph B. Bidwell, assistant head of the Mechanical Development Department, Research Laboratories, described "Engine Bearing Design Today" before the American Society of Lubrication Engineers meeting in Cincinnati, Ohio, on April 6.

On April 10, before the Upper Michigan Medical Society meeting in Petosky, **Calvin H. Hughes**, research biologist in the Industrial Hygiene Department of the Research Laboratories, outlined "The Development of the Mechanical Heart."

On May 11 Mr. Hughes appeared before the Mechanical Engineering Group of the Naval Reserve in Detroit. The title of his presentation was "The Mechanical Heart and its Clinical Application."

Norman A. Hunstad, senior research engineer, and **Richard A. Randall**, research engineer, both of the Fuels and Lubricants Department of the Research Laboratories, addressed a joint session of the American Industrial Hygiene Association and the American Conference of Governmental Industrial Hygienists, meeting in Chicago, Illinois, on April 27. The title of their talk was "The Reliability of Combustible Gas Alarms."

Glen Eastman, sales engineer at Rochester Products Division, appeared before the Toastmasters' Club of Rochester, New York, on May 12. The title of his talk was "Sales Techniques."

R. T. Psik, drafting group engineer in the Product Engineering Department of Ternstedt Division, appeared before U. S. Army Reserve officers at Fort Wayne, Detroit, Michigan, on March 23. The title of his talk was "Amphibious Operation."

On March 25 **Tom C. Stott**, passenger vehicle engineer at Vauxhall Motors, Limited, Luton, England, read and discussed the fifth and final presentation of the paper "Problems in the Design and Development of an Economical Automobile Gearbox" before the Automobile Division of the Institution of Mechanical Engineers in Bristol, England.

Contributors to July-August 1954 Issue

of

GENERAL MOTORS

Engineering

JOURNAL
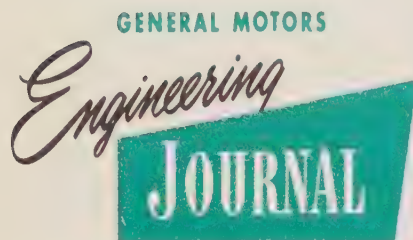
ERIC R. BRATER,

contributor of "An Approach to the Design and Development of a New Product: A Light-Weight, Two-Cycle Diesel Engine," serves as assistant chief engineer of the Engineering Department at Cleveland Diesel Engine Division. His current engineering activity centers around the design and development of Diesel engines —the subject of his paper in this issue.

Mr. Brater's career with Cleveland Diesel Engine began in 1933 when he joined the Division as technical assistant to the director of engineering. Two years later he was promoted to assistant chief engineer, his present position. Mr. Brater was instrumental in organizing the Technical and Metallurgical Departments at Cleveland Diesel Engine to the level required for specialized engineering work as described in his current paper.

After earning the B. S. degree in mechanical engineering from Massachusetts Institute of Technology in 1924, Mr. Brater was employed for nine years by the White Motor Company in Cleveland, Ohio, serving as chief technical engineer for the last four years of his employment. There, he recognized the distinct advantage of concentrating technical and mathematical work in a specialized group of experts, functioning as a separate unit which closely coordinates the actual layout work. This working pattern was later developed and applied at Cleveland Diesel Engine Division.

Mr. Brater twice has presented technical talks on Diesel-engine design and development before the Society of Automotive Engineers and once before the Diesel Engine Manufacturers Association.

Mr. Brater's technical affiliations include membership in the American Society for Metals, the Society for Experimental Stress Analysis, and the Society of Naval Engineers. He also is a member of the Society of Automotive Engineers, serving on the S.A.E. Vibration Committee.

DR. LOUIS J. CANTONI,

contributor of "The New Engineer Has a New Dimension," is a faculty member of the General Motors Institute, serving on the staff of the English and Psychology Department. He has been with G.M.I. since 1951 and he now serves as a teacher of psychology in the cooperative engineering, business-administration, and dealer programs. Other responsibilities include the training of General Motors management personnel in human-relations skills. Prior to his current assignment, he was a conference leader in the Plant Management Training Department.

Dr. Cantoni received his A.B. degree in English and philosophy from the University of California at Berkeley. His master's degree, in social work and his Ph.D. degree, in education and psychology, were granted by the University of Michigan.

He developed his paper on the new engineer after examining the much-printed, facetious definition which he quotes. He felt concerned because this definition failed to incorporate an important element which today's engineer must have—psychological insight into the significance of his own relationships with colleagues and others in a technological world.

Recently, he completed the writing of a detailed, comprehensive unit outline for a program in Marriage and Community Relations. This is now being used as a point of departure in discussion groups comprised of senior cooperative students at G.M.I.

Dr. Cantoni is a member of the American Philosophical Association, the American Personnel and Guidance Association, the American Psychological Association, and the American Society for Engineering Education. He is also a member of the Phi Delta Kappa and the Phi Kappa Phi, honorary scholastic societies.

PAUL FITZPATRICK,

contributor of this issue's "Notes About Inventions and Inventors," has served since December 1949 as a patent attorney in the Patent Section of the Central Office Staff, Detroit.

Mr. Fitzpatrick received his engineering education at Georgia Institute of Technology, earning the B. S. degree in electrical engineering in 1931. In 1941 he received the L.L.B. degree from Georgetown University.

His patent work is related principally to gas turbine engines, automobile bodies, and off-the-road vehicles. He shares the Patent Section's responsibilities of handling patent infringement questions, of protecting inventions originating with General Motors employes, and of negotiating license agreements.

After earning his engineering degree, Mr. Fitzpatrick served for two years with the Navy Department, checking the design of communication systems for warships. Later, during World War II, he wrote technical training material about naval gunfire-control equipment. While studying for his L.L.B. degree he was employed as an examiner in the United States Patent Office, Washington, D. C.

Mr. Fitzpatrick was a patent solicitor with a private law firm in Cleveland, Ohio, from 1945 to 1949. He is a registered patent attorney, a member of the bar of the District of Columbia, and a member of the Michigan Patent Law Association.

JAMES F. HAGEN,

co-contributor of "Design Considerations Applying to Specification of Surface Finish for Machined Parts" and "A Discussion of Instrumentation for Determining Surface Roughness of Machined

Parts," is a research engineer in the Research Laboratory Division's Mechanical Development Department, located at the General Motors Technical Center.

Work on the development of precision reference specimens of surface roughness, the project on which Mr. Hagen is currently engaged, prompted the incorporation of some of these findings in his two papers for the GENERAL MOTORS ENGINEERING JOURNAL.

Mr. Hagen, who first joined General Motors in August 1948 in the Research Laboratories Division, began as a college graduate-in-training attached to the Personnel Department of the Research Laboratories. During his year in training, he worked in the Mechanical Development, Automotive Engines, and Physics and Instrumentation Departments. In 1949 he was made junior engineer and two years later, in August 1951, was advanced to his present position of research engineer.

He received his formal education at the University of Colorado at Boulder where he was granted the Bachelor of Science degree in mechanical engineering in 1948.

Mr. Hagen is affiliated with Tau Beta Pi, Pi Tau Sigma, Sigma Pi Sigma, and Pi Mu Epsilon, honorary societies.

WALTER F. HELLER,

co-contributor of "Helium-Nitrogen Shielding Gas Improves Welding of Low-Carbon Steel," is director of quality control at the Kansas City plant of the Buick-Oldsmobile-Pontiac Assembly Division. Mr. Heller joined this Division, which is set up to produce both automobiles and aircraft simultaneously, in 1951 as chief metallurgist and manufacturing development engineer on the aircraft program. In 1953 he was promoted to his present position and is currently engaged in developing aircraft test equipment, manufacturing processes, and inspection procedures.

Mr. Heller received the B.S. degree in electrical engineering in 1926 from Clemson College and later did graduate work in metallurgical and mechanical engineering. In 1928 he joined AC Spark Plug Division, Flint, Michigan, where he

remained until 1944, attaining the position of chief metallurgist. While at AC Spark Plug, Mr. Heller was associated with the Process Engineering, Material and Production Control, Manufacturing, and Laboratory Departments. He has worked on varied process, equipment, and product projects, with outstanding developments in automotive and aircraft bearings and spark plugs.

Mr. Heller's engineering experience outside of General Motors includes work with the Michigan Bell Telephone Company, the Bethlehem Steel Corporation, and the Nelson Specialty Welding Equipment Company where he was technical director and executive assistant to the president.

Five patents in the fields of engineering design and metallurgy have resulted from Mr. Heller's work. His technical affiliations include membership in the American Welding Society and the American Society for Metals.

WALTER B. HERNDON,

contributor of "An Application of Hydraulic Fundamentals: Development of the Hydra-Matic Automatic Transmission," serves as works manager at the Detroit Transmission Division, Willow Run, Michigan.

Mr. Herndon began his engineering career with General Motors in July 1928 as a tool designer at the Cadillac Motor Car Division in Detroit. He later transferred to the Cadillac Engineering Department as a designer. He continued in this capacity until 1939 when he transferred to the Central Office Engineering Staff as a project engineer. When Detroit Transmission Division was formed in the same year, he transferred to the new Division's Engineering Department and later was made assistant to the general manager. Two years later he became assistant chief engineer at the same Division, serving in this position until 1949 when he was promoted to chief engineer. In 1952 he was appointed director of engineering and sales and one year later was named works manager. In this position, he supervised the establishment of manufacturing facilities at the Willow Run plant when the Division moved in 1953.

Earlier in his engineering career, Mr.

Herndon figured prominently in the initial design work and production of the Hydra-Matic automatic transmission, the subject of his current GENERAL MOTORS ENGINEERING JOURNAL paper. His work in the field of automatic transmissions has resulted in the granting of six patents with six others still pending.

Mr. Herndon graduated in 1928 from the State College of Washington with the B.S.E. degree. He received the M.S.E. degree in 1930 from the University of Michigan.

He is a member of the Engineering Society of Detroit and the Society of Automotive Engineers, the latter having published his paper on "The Hydra-Matic Transmission" in the January 1952 S.A.E. Quarterly Transactions.

WILBUR F. KARBER,

contributor of "How Stress Problems Are Anticipated and Solved in Automotive Bodies," is a stress engineer who has made a transition from designing steel and concrete buildings to aircraft-structure stress analysis and, ultimately, to automotive-body stress analysis and structural design. He currently is a senior stress engineer in the Product Engineering Department of Fisher Body Division, General Offices Plant, Detroit, Michigan. In this capacity he is concerned with stress analysis and structural-design work on new body styles.

Mr. Karber originally joined Fisher Body at this Division's Cleveland, Ohio, plant in March 1943 as a stress engineer on the B-29 aircraft program. In September of that year he was promoted to supervisor of stress engineers. In 1946 Mr. Karber transferred to the General Offices Plant and assumed his present position.

Before joining Fisher Body Division, Mr. Karber had 17 years' experience as a civil engineer starting in 1925 after receiving the B.S. degree in civil engineering from the Case School of Applied Science (now Case Institute of Technology). After graduation, Mr. Karber continued his education by taking extensive night-school courses dealing with intermediate-structures design. His work as a civil engineer was devoted mainly to the design of municipal steel and concrete structures. His experience included employment in the Utilities Department

of the City of Cleveland where he worked on the design and construction of water-treatment plants.

JOHN J. KILMER,

contributor of "Flexible Design of Power Plant Provides Needed Power Now with Minimum Disruption of Overall Plan," serves as chief engineer of power houses at AC Spark Plug Division, Flint, Michigan. In this capacity he is in charge of this Division's three power houses in the Flint area.

Mr. Kilmer originally joined General Motors in 1920 when he was employed by the Buick Motor Division as general foreman of electricians. In 1925 he transferred to AC Spark Plug as head of the Meter Testing Department and was responsible for the testing and maintenance of all recorders and controllers used in the plant's manufacturing processes.

In 1932 Mr. Kilmer was promoted to chief electrician and in this capacity had charge of all electrical maintenance and construction work. In 1939 he was promoted to his present position. Since that time, the overall steam output of the three power houses has increased $5\frac{1}{2}$ times. One of his earliest projects was supervising the construction of the Industrial Avenue power house of AC Spark Plug which was needed as a result of expanding manufacturing facilities. In the design and construction of the new power house of which he writes in this issue, Mr. Kilmer supervised the selection of equipment, placement, and overall layout.

Mr. Kilmer was graduated from the University of Michigan in 1918 with the B.S. degree in electrical engineering and saw service in the Army Signal Corps prior to joining General Motors.

EARL E. LINDBERG,

co-contributor of "Design Considerations Applying to Specification of Surface Finish for Machined Parts" and "A Discussion of Instrumentation for Determining Surface Roughness of Machined Parts," is a research engineer in the

Mechanical Development Department of the Research Laboratories Division, located at the General Motors Technical Center.

The Mechanical Development Department is concerned with a variety of projects such as unconventional engines, Diesel engines, fatigue testing, friction, lubricants, and bearings. A considerable portion of Mr. Lindberg's work has been related to the development of electronic instrumentation for this Department.

His employment with General Motors began in August 1951 when he started in the Personnel Department of the Research Laboratories Division as a college graduate-in-training. By June of the next year he was made a junior engineer in the Mechanical Development Department. Six months later he began his present work.

Mr. Lindberg received the Bachelor of Science degree in electrical engineering from the University of Nebraska in 1949.

His military-service record includes over three years in the United States Navy as a chief radio technician during World War II.

ANTHONY J. PANE,

who prepared the typical problem in automotive design "Determine the Tensile Stress in Pump-Cover Bolts on a Torque-Converter Transmission" and the solution appearing in this issue, serves as a project engineer in the Engineering Department of the Buick Motor Division, Flint, Michigan. In this capacity, Mr. Pane's current work is concerned with torque-converter pump-test analysis.

Mr. Pane was graduated from the University of Michigan in 1942 with the B.S. degree in mechanical engineering. Prior to his entrance into the University, he was the recipient of an Industrial Union Scholarship and a Bausch and Lomb Science Award, both from the Dunkirk, New York, High School. Shortly after his graduation from the University of Michigan, Mr. Pane entered military service and attended Officer Candidate School, from which he was graduated with the rank of lieutenant in the Army Corps of Engineers. In this capacity, he was connected with the Manhattan

Project and served as an atomic-bomb assembly officer.

In 1948 Mr. Pane joined Buick as a laboratory assistant in the Dynamometer Department. One year later he entered into a student-engineer training program and after completion of the program assumed his present position.

Mr. Pane's technical affiliations include membership in the Society of Automotive Engineers.

EUGENE L. TURNER,

co-contributor of "Helium-Nitrogen Shielding Gas Improves Welding of Low-Carbon Steel," is a welding engineer at the Kansas City plant of the Buick-Oldsmobile-Pontiac Assembly Division. The subject of his paper is the result of an interest in the technological aspects of welding which he developed while working as a construction and armour-plate welder and as a service engineer for the National Cylinder Gas Company, prior to joining the Kansas City B.O.P. Assembly Division in 1951. His interest led him to experiment with various welding-gas mixtures which would produce a better weld at a highly reduced cost. His continued work in this area after joining the Division aided considerably in developing the application of helium-nitrogen gas to welding.

Mr. Turner is a member of the American Welding Society and a veteran of World War II. He served three years in the Pacific War Theater and was discharged with the rank of master sergeant.

When Mr. Turner became associated with General Motors' new dual-purpose plant in Kansas City, he was assigned the project of coordinating fusion welding for the new aircraft program.

He coordinated all welding specifications with the F-84F designers, set up the fusion-welding procedures for equipment and operations, and assisted in the development of a training program for Air Force certification of fusion welders.
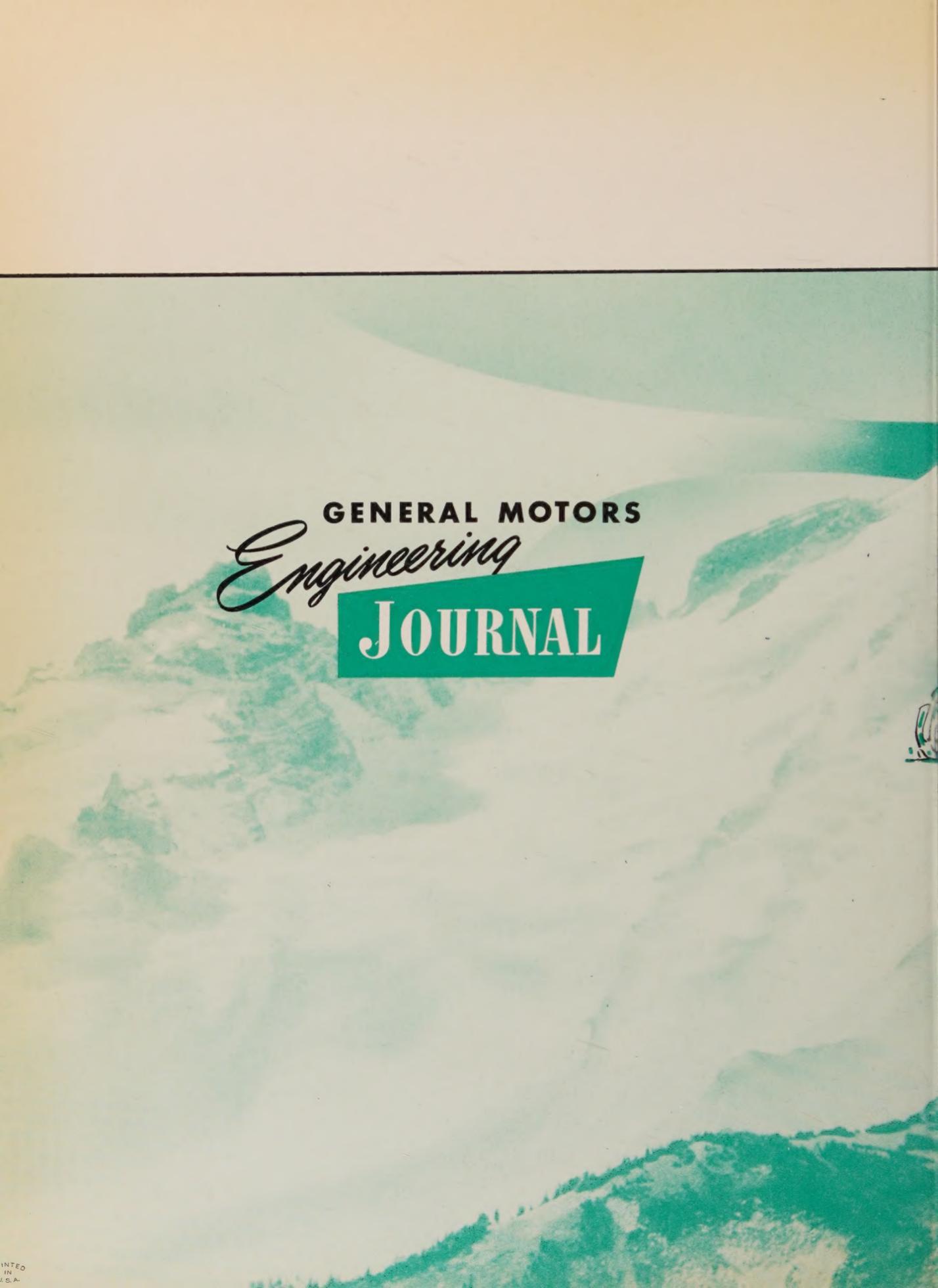
His previous projects at the Kansas City B.O.P. Assembly Division have included the writing of a welding training manual. He is currently engaged in research and development in the field of fusion welding.
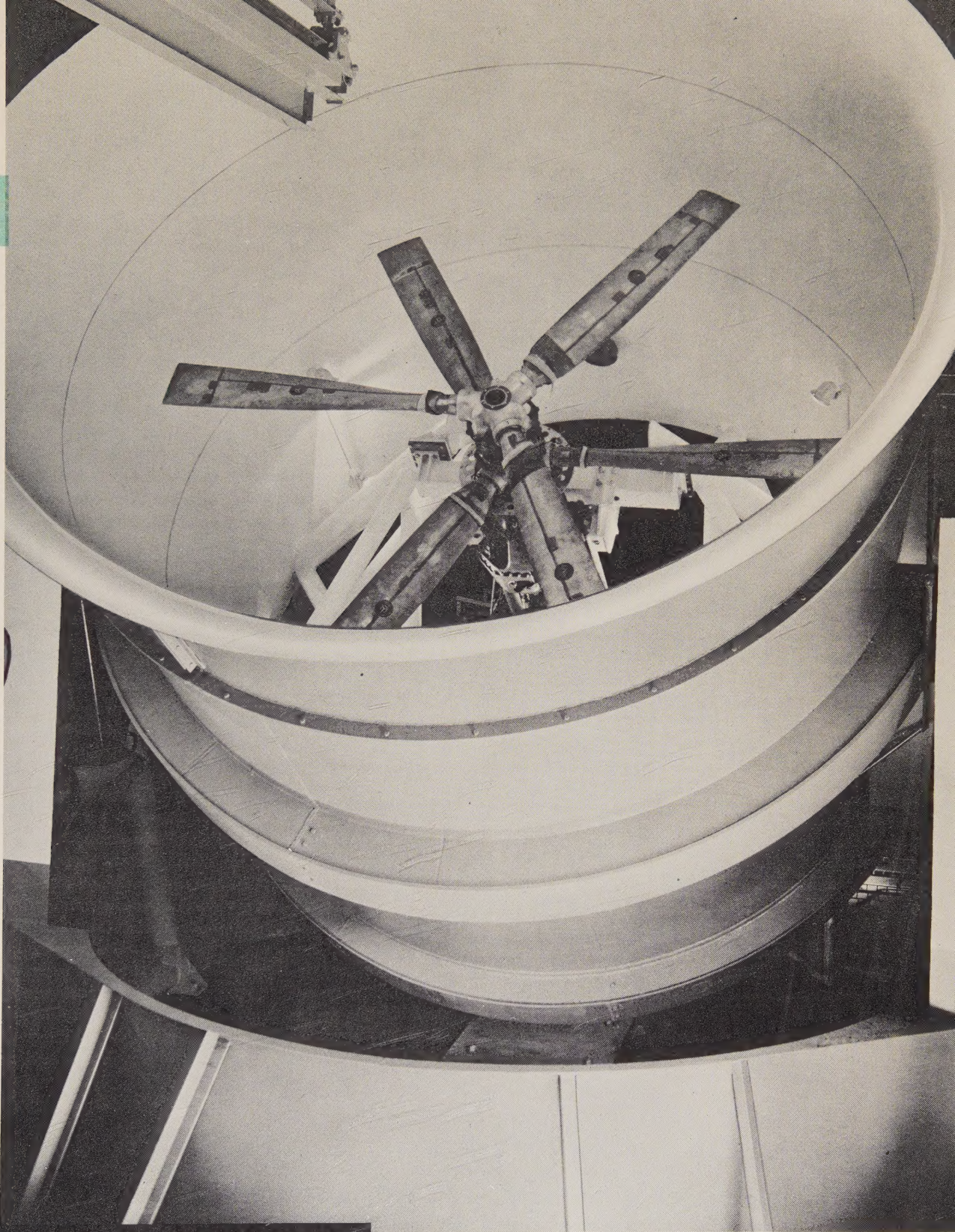
VERTICAL ENGINEERING

In adapting Allison Division's T40 turbo-prop engine for use in *VTO* (vertical take-off) aircraft, engineers devised a unique engine test stand. Tiltable from horizontal to vertical positions, the tunnel-shaped stand enabled study of and satisfactory solution of the unusual problems associated with lubrication, fueling, and general functioning of an engine and propeller in a nose-up attitude. The large propeller required a 17-ft diameter tunnel.

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